LBNL-59998

ERNEST ORLANDO LAWRENCE BERKELEY NATIONAL LABORATORY

Evaluation of Existing Technologies for Meeting Residential Ventilation Requirements

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July 2006

Residential Ventilation Requirements and Distribution System Research for 2008 Building Efficiency Standards

Task 3.2 Evaluation of Existing Technologies for Meeting Residential Ventilation Requirements

Prepared in Support of

LBNL Technical Assistance for PIER

Contract Number 500-04-005

This work was supported by the Assistant Secretary for Energy Efficiency and Renewable Energy, Office of the Building Technologies Program, U.S. Department of Energy under Contract No. DE-AC02-05CH11231.

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Introduction

This report summarizes the evaluation of existing technologies for meeting residential ventilation requirements for potential changes to California Building Energy Efficiency Standards (commonly referred to as Title 24). This evaluation was performed by simulating a range of ventilation systems in California climates. The first part of this work was to develop a simulation plan. This plan was reviewed by commission staff and the project PAC. This report summarizes the simulation plan as well as the simulation results. More detailed information including summary spreadsheets and files of hourly performance for each simulation will be made available to the commission separate from this report.

Simulation Plan

This simulation plan outlines the simulations we carried out to investigate the energy and IAQ implications of different technical approaches to meeting potential Title 24 ventilation requirements. The information required to simulate each approach is summarized together with rationales for selection of particular parameters. The technologies are discussed in more detail in the companion Literature Review[1].

To determine the energy used to provide mechanical ventilation, we used the HVI Directory[2] to obtain fan power for fans that met the air flow requirements proposed for Title 24 and the sound requirements of ASHRAE Standard 62.2. In this plan, the specific fan manufacturers and model numbers are given in square parentheses [] for each system.

Approximately 100 different combinations of house size, climate and ventilation technologies were simulated. We used the REGCAP[3] simulation model that performs minute-by-minute simulations and produces hourly data for post-processing. The REGCAP model has been used in several previous studies of HVAC system performance[4]. REGCAP has a detailed air flow network model that calculates the air flow through building components as they change with weather conditions and HVAC system operation. The pressure difference and airflow calculations include the effects of weather, leak location, and HVAC system flows on house and attic air pressures. These dynamic air pressure and air flow interactions are particularly important because the air flows associated with ventilation systems (including duct leakage) significantly affect natural infiltration in houses.

Houses to be simulated

Three house sizes were simulated to examine the implicit effect of occupant density in the 62.2 requirements. Because the number of occupants does not scale with the size of a house larger houses tend to have lower occupant densities. For most of the mechanical

[1] McWilliams and Sherman. 2005. Review of Literature Related to Residential Ventilation Requirements. LBNL 57326.
[2] HVI. 2005. Certified Home Ventilating Products Directory, Home Ventilating Institute.
[3] The appendix gives details of the simulation model.
[4] See REGCAP Bibliography at the end of the Appendix.

ventilation simulations, the medium sized house was used, and for selected cases we will use the smaller and larger houses.

1. Small (1,000 ft^2) one-story two-bedroom house.
2. Mid-size (1,761 ft^2) two-story, three-bedroom house[5]
3. Large (4,000 ft^2) two-story, five-bedroom house.

Envelope leakage for each house was fixed with an SLA of 4 as this was considered by the PAC to be a reasonable value for new California construction. The corresponding leakage values are summarized in Table 1.

House and duct insulation used to determine the non-ventilation building load and duct system performance will vary by climate as shown in Table 2[6]. The insulation is degraded according to the 2005 Residential Alternative Compliance Manual[7].

Exterior surface area for wall insulation scales with floor area and number of stories. A simple rule of thumb developed from measured data from several thousand new homes[8] and from the simplified box prototype C in the ACM is that the wall area is typically 1.54 times the floor area for a two-story home and 1.22 times the floor area for a one-story home. Window area is 20% of floor area with windows equally distributed on the four exterior walls. The SHGC varied by climate zone between 0.4 and 0.65. Values specified in T-24 Table 151-C, p.133 were used. In climate zones where a minimum SHGC was not required, T-24 Table 116-A and 116-B, p.56 were used. The required U-value was found in Table 116-A, and then the SHGC corresponding to the same window from table 116-B was used. Clear glazing was assumed together with an exterior shading of 50%.

Table 1. Envelope Leakage

Floor Area (ft^2)	SLA	ELA_4 (in^2)	$m^3/(sPa^n)$	cfm/Pa^n
1,000	4	58	0.038	81
1,761	4	101	0.067	143
4,000	4	230	0.152	325

[5] Based on the 2005 T24 ACM prototype C
[6] Based on CA T24 2005 Package D requirements including degradation factors.
[7] California Energy Commission. 2005.
[8] Based on BSC/Building America data

Table 2. House Insulation Levels						
Climate Zone	Ceiling			Wall		Ducts outside conditioned space
		Heating Degraded	Cooling Degraded		Degraded	
1	R38	21.6	31.9	R21	17.6	R6
2	R30	18.8	26.1	R13	10.9	R6
3	R30	18.8	26.1	R13	10.9	R6
4	R30	18.8	26.1	R13	10.9	R6
5	R30	18.8	26.1	R13	10.9	R6
6	R30	18.8	26.1	R13	10.9	R4.2
7	R30	18.8	26.1	R13	10.9	R4.2
8	R30	18.8	26.1	R13	10.9	R4.2
9	R30	18.8	26.1	R13	10.9	R6
10	R30	18.8	26.1	R13	10.9	R6
11	R38	21.6	31.9	R19	10.9	R6
12	R38	21.6	31.9	R19	10.9	R6
13	R38	21.6	31.9	R19	10.9	R6
14	R38	21.6	31.9	R21	17.6	R8
15	R38	21.6	31.9	R21	17.6	R8
16	R38	21.6	31.9	R21	17.6	R8

Meeting Proposed Ventilation Requirements

The proposed requirements are to have mechanical ventilation that meets ASHRAE Standard 62.2 plus an extra 25 cfm of capacity to allow for periodic turning off of the system (whether as part of a controlled ventilation system or by occupant intervention).

Whole Building Ventilation

For ASHRAE 62.2, mechanical ventilation is sized as follows:

$$Q(cfm) = 0.01A_{floor}\left(ft^2\right) + 7.5(N+1)$$
$$Q(L/s) = 0.05A_{floor}\left(m^2\right) + 3.5(N+1) \qquad \textbf{(1)}$$

where N is the number of bedrooms in the house.

For the three house sizes we plan to simulate:
1000 ft^2 & 2 bedrooms (3 occupants) $\Rightarrow$ 33 cfm
1761 ft^2 & 3 bedrooms (4 occupants) $\Rightarrow$ 48 cfm
4000 ft2 & 5 bedrooms (6 occupants) $\Rightarrow$ 85 cfm

Adding the extra 25 cfm results in:

1000 ft^2 & 2 bedrooms (3 occupants) $\Rightarrow$ 58 cfm
1761 ft^2 & 3 bedrooms (4 occupants) $\Rightarrow$ 73 cfm
4000 ft2 & 5 bedrooms (6 occupants) $\Rightarrow$ 110 cfm

Using continuous operation of bathroom exhaust requires a minimum of 20 cfm (From 62.2 Table 5.2), and all of these proposed systems exceed this minimum.

Intermittent Operation

Intermittent exhaust was simulated as a peak demand reduction technique (and possibly outdoor pollutant control). The system consists of a bathroom fan that is on for 20 hours and off for 4 hours during peak (3-7 p.m. for cooling and 1 – 5:00 a.m. for heating). The relationships given in Sherman (2005)[9] and in ASHRAE 62.2 show that intermittently under ventilating for 4 hours out of 24 (given the background natural infiltration and extra 25 cfm capacity of the continuous exhaust minimum flow required by 62.2) gives acceptable effective ventilation rates that meet 62.2 requirements.

Additional 62.2 requirements

All the fans used to provide mechanical ventilation were selected to meet the sound and installation requirements of 62.2. From an energy use perspective, the main effect is that fans that meet the 1.0 Sone requirement for continuous operation and 3 Sones for intermittent operation tend to be energy efficient fans that also have power ratings in the HVI directory[10].

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[9] Sherman, M.H. 2005. "Efficacy of Intermittent Ventilation for Providing Acceptable Indoor Air Quality", ASHRAE Transactions, Vol. 112., ASHRAE, Atlanta, GA.
[10] HVI. 2005. Certified Home Ventilating Products Directory, Home Ventilating Institute.

Weather

We will use Title 24 compliance hourly data files converted to minute-by-minute format by linear interpolation. The simulations also use location data (altitude and latitude) in solar and air density calculations. The required weather data for the simulations are:

- direct solar radiation (W/m^2)
- total horizontal solar radiation (W/m^2)
- outdoor air dry-bulb temperature(°C)
- outdoor air humidity ratio
- wind speed (m/s)
- wind direction (degrees)
- barometric pressure (kPa)
- cloud cover index

Heating and Cooling Equipment

The simulations used the detailed equipment models discussed in more detail in Appendix A. Equipment sizing was based on a combination of Manual J calculations and the results of the field survey of new California homes being undertaken by Rick Chitwood[11]. Equipment sizing is most important when considering systems that use the central furnace blower to distribute ventilation air because the outside air is usually supplied as a fraction of total furnace blower flow and the energy used to distribute the air depends on the size of the blower motor (Appendix E summarizes the heating/cooling equipment capcities and associated blower power consumption). For all these simulations, the correct furnace blower flow and refrigerant charge were used, so air conditioner capacity and EER will only depend on the return air and outdoor air temperatures.

The heating was supplied by an 80% AFUE natural gas furnace. For cooling, a SEER 13 split-system air conditioner with a TXV refrigerant flow control was used.

The duct leakage to outside was 5%, split with 2.5% supply leakage and 2.5% return leakage for most of the simulations. A few cases were examined with higher duct leakage: 11% supply and 11% return[12].

[11] PIER 08 Residential Furnace blower Survey – see results summary in Appendix C
[12] Title 24 default for new construction

Determination of heating or cooling operation was based on the Title 24 seven day running average technique. When the seven day running average outdoor temperature is greater than 60°F then we have cooling and if it is less than 60°F we have heating. However, in most climates this results in multiple switches between heating and cooling that is unrealistic. Therefore, for each climate zone, we will select one day for the heating to cooling mode switch and one day for the cooling to heating mode switch based on the seven day running average technique. A list of the switching days is given in Table 3.

Table 3. Days to switch heating and cooling modes

CZ	Day to switch to cooling	Day to switch to heating
1	No cooling	Always in heating mode
2	134	289
3	152	283
4	152	284
5	185	286
6	144	310
7	115	310
8	108	313
9	112	313
10	113	313
11	117	282
12	117	278
13	103	300
14	133	289
15	64	317
16	160	247

Operation of the heating and cooling equipment used the following set-up and set-back thermostat settings taken from the Residential Alternative Calculation Method (ACM) Approval Manual for the 2005 Building Energy Efficiency Standards for California.

Table 4. Thermostat Settings for Ventilation Simulations (°F)

Hour	Heating	Cooling
1	65	78
2	65	78
3	65	78
4	65	78
5	65	78
6	65	78
7	65	78
8	68	83
9	68	83
10	68	83
11	68	83
12	68	83
13	68	83
14	68	82
15	68	81
16	68	80
17	68	79
18	68	78
19	68	78
20	68	78
21	68	78
22	68	78
23	68	78
24	65	78

Ventilation Technologies to be Simulated

1. Unvented House

This case represents a California home built to comply with 2005 Title 24 building and energy codes, but that does not comply with ASHRAE Standard 62.2 and does not have the ventilation adder used in Title 24 (this was considered in separate simulations). We simulated all 16 climate zones for the medium house. The envelope leakage was the same as the mechanically ventilated homes.

2. Continuous exhaust

The air flow requirements were met using envelope infiltration and continuous exhaust through a bathroom fan. The medium sized house was simulated in 16 climate zones; the small and large house were simulated in five climate zones (3, 13, 16, 15, &10). These climate zones were chosen as they contain the majority of new construction in the state.

The ASHRAE 62.2 requirements are:

- 1000 ft^2 & 2 bedrooms (3 occupants) ⇒ 58 cfm
- 1761 ft^2 & 3 bedrooms (4 occupants) ⇒ 73 cfm
- 4000 ft2 & 5 bedrooms (6 occupants) ⇒ 110 cfm

Using the nearest size greater than the minimum using specific directory entries gives the following for fan power use:

1000 ft^2 & 2 bedrooms (3 occupants) ⇒ 60 cfm [0.028 m3/s] 13.7 W [Panasonic FV-05VQ2]
1761 ft^2 & 3 bedrooms (4 occupants) ⇒ 73 cfm [0.034 m3/s] 20.1 W [Panasonic FV-08VQ2]
4000 ft2 & 5 bedrooms (6 occupants) ⇒ 60 cfm [0.028 m3/s] 13.7 W [Panasonic FV-05VQ2] + 50 cfm [0.0236 m3/s] 13.5 W [Panasonic FV05VF1] (Total of 27.2 W)

The baseline for comparing ventilation technologies was the medium-sized 1761 ft^2 house with continuously operating exhaust.

3. Intermittent exhaust

The simulations were performed for all three house sizes in a heating dominated (CZ 16), in a cooling dominated (CZ 13) climate, and in a temperate climate (CZ3). The fan flow and power requirements are the same as for case 2. Note that the air flow rates and equipment for case 2 can be used in this case because 25 cfm was already added to the 62.2 minimum for case 2.

4. Heat Recovery Ventilator (HRV)

Typical HRV installations do not operate continuously. In these simulations, the operated for half an hour then were off for half an hour. The HRV air flows in the HVI directory are typically much larger than the minimum 62.2 requiremnts. We chose one of the lowest air flwo HRV with an air flow of 130 cfm. This is about 35% more flow than simply doubling the 62.2 minimum requirement of 48 cfm for this house that would be required for its 50% duty cycle.

An HRV was simulated for the medium sized house in cold climates (CZ 16 and CZ 1). The HVI listed recovery efficiencies were applied to the air flow through the HRV when calculating the energy use. For these simulations, the Apparent Sensible Effectiveness (ASE) was used to determine the temperature of air supplied to the space ($T_{tospace}$). It was assumed that the HRV has its own duct system that does not leak and is located entirely within the conditioned envelope of the house.

$$ASE = \frac{T_{out} - T_{tospace}}{T_{out} - T_{fromspace}}$$

The following HRV was selected from the HVI directory:
[Broan Guardian HRV 100H]. At 138 cfm [0.0652 m^3/s] net airflow at the 0.44 inches of water [110 Pa] external static pressure of the standard HVI rating point (we assumed that the HRV was installed correctly and has this rated pressure drop), it uses 124 W and has:

- Apparent Sensible effectiveness = 70%
- Sensible recovery efficiency = 62%

It was assumed that the supply and return fans used the same amount of power, i.e., 62W. For the supply fan 55W of heat was added to the internal (based on 7W of required air power).

An addition set of simulations was performed at a reduced operating schedule (35% less operating time) such that the mean ventilation rate was the same as the 62.2 minimum requirements. To distinguish between the two HRV schedules, the ones with a mean rate matching the 62.2 minimum are case 4 and those on a 50% duty cycle are 4X.

5. Central Fan Integrated (CFI) Supply with air inlet in return and continuously operating exhaust

CFI and continuous exhaust was simulated for all three houses in CZs 3,13,16,15,and 10. The continuously operating exhaust performance is the same as case 2. This is augmented with a central fan integrated supply that uses the furnace blower to intentionally draw outdoor air through a duct into the return and distribute it throughout the house using the heating/cooling supply ducts. The outdoor air duct is only open to outdoors during furnace blower operation and has a damper that closes when the furnace blower is off. This damper was assumed to have zero leakage when closed.

The furnace fan power requirements were determined based on the space conditioning equipment capacity determined by Manual J load calculations and a nominal 2 cfm/W (that has been found to be typical in numerous field studies). Because the CFI systems used the forced air heating and cooling ducts, the same air leakage and heat transfer was applied to the ducts for CFI operation as for heating and cooling operation. For this study, it is assumed that ducts are in the attic. The waste heat from the furnace blower and heat exchange between the ducts and their surroundings were included in the calculations. The fraction of outside air (OA) entering the system is fixed so that it balances the exhaust flow and makes this system switch from exhaust ventilation to balanced ventilation. The central fan integrated supply system operated for at least 20 minutes per hour if the heating and cooling systems operate for less than this time to satisfy thermostat calls for heating or cooling.

To examine sensitivity to duct leakage, we simulated the medium sized house in a heating dominated (CZ 16), cooling dominated (CZ 13) and temperate climate (CZ3) with 11% supply an 11% return leakage.

6. Continuous Supply

We simulated the medium house in CZs 3,13,16,15,and 10. The continuous supply system will use a fan to supply filtered air from outside that then distributes the air throughout the house without using the furnace blower or the forced air heating and cooling ducts. Therefore the continuous supply air is not associated with any duct leakage or heat transfer effects. For continuous supply, the supply air is mixed with indoor air for tempering purposes. We will use a mixing ratio of 3:1 for indoor to supply air. The supply fan will therefore be sized to be four times the case 2 requirements, i.e., 292 cfm [0.138 m^3/s] for the medium sized house. A [Greentek MTF 150P] provides this flow at a power consumption of 133 W of which 14 W is air power and 119 W is heat.

Because this supply fan will normally be an inline fan located outside the building thermal envelope, an exception in 62.2 means that it does not have to meet the low Sone requirement. This is fortunate, as the inline fans in the HVI directory either do not have sone ratings or do not meet the low sone requirements in 62.2.

7. CFI with 7% Outside Air (OA), without continuous exhaust – *not 62.2 compliant*

These simulations were performed for the medium house in CZs 3, 10 ,13, 15 and 16. Unlike the case 5 simulations, there was no continuous exhaust. The CFI system was on for 10 minutes, then off for 20 minutes. This system does not account for furnace blower operation for heating or cooling: the outdoor air supply duct was open and the the blower was on for the first 10 minutes out of every 30 minutes regardless of the space conditioning system operating mode. Because the air flows from outside are limited by tempering issues they are the same as for case 5. Due to reduced operating time the net flows are therefore not 62.2 compliant. The outdoor air flow is based on the total furnace blower flow and was set at 7% of fan flow. Because this is achieved by a fixed damper setting rather than damper modulation to achieve a fixed flow, this air flow is a fixed 7% of the furnace blower flow. I.e., 7% of heating fan flow during heating, 7% of cooling fan flow during cooling and 7% of cooling fan flow when ventilating only). A damper closes the outside air vent when the CFI is not operating (i.e. for 20 minutes out of every 30 minutes).

8. CFI with 1/3 of 62.2 flow, without continuous exhaust– *not 62.2 compliant*

These simulations are the same as case 7 but with the air flow adjusted to be the 62.2 air flow rate rather than 7% of blower flow. Because the CFI operates one third of the time it provides one third of the 62.2 required air flow.

9. Minimum Ventilation from ACM

These simulations were for the unvented house of case 1, but with the minimum ventilation rate adder of 0.35 ACH used when air change rates fall below 0.35 ACH. This mimics the ventilation added currently used in the Title 24 ACM.

Source Control Ventilation

In addition to the specific technologies that meet 62.2, we will include intermittent operation of kitchen and bathroom fans.

Intermittent bathroom fans will operate for half an hour every morning from 7:30 a.m. to 8:00 a.m. These bathroom fans were sized to meet the 62.2 requirements for intermittent bathroom fans. From Table 5.1 in 62.2 this is 50 cfm (25 L/s) per bathroom. For houses with multiple bathrooms, we will assume that the bathroom fans are operating at the same time, so the 1,761 ft^2 house will have a total of 100 cfm (50 L/s) and the 4,000 ft^2 house will have a total of 150 cfm (75 L/s). Power requirements for these fans are 0.9 cfm/W based on the Chitwood field survey data, i.e. 55W for each 50 cfm fan.

Similarly, all simulations will have some kitchen fan operation. Based on input from ASHRAE Standard 62.2 members and an ARTI project monitoring committee, the kitchen fans will operate for one hour per day from 5 p.m. to 6 p.m. These kitchen fans were sized to meet the 62.2 requirements for intermittent kitchen fans. From Table 5.1 in 62.2 this is 100 cfm (50 L/s). Unfortunately, very few of the kitchen fans in the HVI directory have power consumption information. The smallest of those that do [Ventamatic Nuvent RH160] has a flow rate of 160 cfm, and uses 99W.

Ventilation Options not simulated

Two ventilation options that we had initially considered simulating (open windows and passive vents) were not simulated for the following reasons:.

- *Open windows*. Based on recent survey results[13], this method of providing ventilation is not sufficiently reliable due to uncontrollable variations in occupant behavior.

- *Passive vents*. Although popular in Europe, this technology is not available in the California market.

This reduction in scope allowed us to perform additional calculations for the other cases: indoor concentrations at low ventilation rates and the effects of the low ventilation rate added in Title 24 ACM (Case 9).

Other ventilation related options not included are:

- *Complex control strategies for any kind of ventilation system*. These are generally not appropriate for a minimum performance standard and it is too difficult to ensure that the actual operating characteristics are what is claimed in compliance calculations. They are also very complex to deal with for compliance software.

- *Proprietary Systems, e.g., Nightbreeze* –proprietary control and operation algorithms are unavailable.

[13] *Price, P.N. and M.H. Sherman* **"Ventilation Behavior and Household Characteristics in New California Houses,"** April 2006. LBNL-59620.

Results of Ventilation Simulations

The following results are for the medium house except where noted.

Air change rates

The air flows were converted into air change rates by dividing by the house volume. Mean annual air change rates were calculated for each simulation and are summarized in Table 5. Effective air change rates were calculated using the Sherman and Wilson[14] turnover time approach that accounts for temporal variation in air change to calculate the effective air change that would give the same internal exposure to pollutants, and are summarized in Table 6. In general, these rates are lower than the mean air change rates. However, as the efficacy (ratio of Effective to mean ACH) values in Table 7 show, the mechanical ventilation systems have about 5 percentage points more effectiveness than the unvented house. This serves to make the differences in air change rates larger between the unvented and mechanically vented house.

Table 5 shows that the mean air change rates when mechanical systems are used have much less CZ to CZ variability compared to non-mechanically ventilated cases. All the 62.2 compliant systems (2, 3, 4, 4X, 5, and 6) have mean air change rates of 0.35 or higher. For the unvented house, it's sensitivity to weather conditions results in a large range of average ventilation rates from 0.19 to 0.32 ACH depending on climate. This variability is mostly driven by cold winter weather that results in higher stack pressures and envelope air flows. All of the mean ventilation rates for the unvented house are lower than any of the 62.2 compliant cases (2 through 6).

Comparing continuous exhaust to the unvented house, the mean effective ventilation rate increased about 65%, with ACH increases ranging from 0.11 ACH in CZ16 to 0.18 ACH in CZ15. The houses in the colder climates have smaller changes when adding mechanical ventilation because they have more natural infiltration. For example, in CZ16, ACH rates are more than 0.5 ACH for the unvented house in the winter. In contrast, for CZ 8, the continuous exhaust increased ventilation rates by over 85% due to low natural infiltration driving forces, even in winter.

Comparing intermittent exhaust to continuous exhaust, the average ventilation rate decreased about 7% due to the 17% reduction in operating hours for the exhaust fan. The ventilation effectiveness changes by less than 1% (and is high at around 97% to 98%) by going to this intermittent strategy. This is because infiltration still occurs when the mechanical ventilation is off, and indicates that the off period of four hours is not too long (and the 24 hour cycling period is short enough).

Comparing HRV to continuous exhaust, the average ventilation rate increased about 45%. This is because the HRV flow is about 35% higher than required to meet 62.2 even

[14] Sherman, M.H. and D.J. Wilson, "Relating Actual and Effective Ventilation in Determining Indoor Air Quality," Building and Environment 21 (3/4): 135-144, 1986. LBL-20424.

when only operating for 30 minutes out of each hour. Reducing HRV runtime to 20 minutes out of each hour resulted in a mean effective ventilation rate of 5% less than continuous exhaust even though the 20 minutes out of every hour is only one third of the operating time required to match the continuous exhaust. This is because the balanced ventilation of the HRV added directly to the natural infiltration, whereas the exhaust only fan only adds about half of its flow to the effective ventilation rate.

Comparing Central Fan Integrated (CFI) Supply with air inlet in return and continuously operating exhaust to continuous exhaust, the average ventilation rate increased by 22% indicating that the added outdoor air supply through the return inlet (with same flow as exhaust) is effective at increasing effective ventilation. This is because when both the CFI and exhaust are operating the system is balanced and balanced systems add directly to the natural infiltration unlike exhaust only systems.

Comparing continuous supply to continuous exhaust, the average ventilation rate increased by 17% indicating that the supply fan is more effective. This is due to a combination of two factors. Firstly, periods of balanced mechanical ventilation when the kitchen and bath exhausts operate (when the exhaust ventilation fan provides even more unbalanced exhaust ventilation). Also, under normal natural ventilation conditions, the leakage distribution and wind and stack effect pressures tend to make the house slightly depressurized that allows supply systems to be slightly more effective in the their interaction with the building envelope.

Two non 62.2 compliant technologies were evaluated because they are currently used as mechanical ventilation systems in new California houses. They are both Central Fan Integrated (CFI) systems that operate for 20 minutes out of each hour. Although their air flow rates for outside air are close to or equal to the 62.2 specified continuous air flow rates their fraction runtime makes them non-62.2 compliant.

The first of these systems has 7% outdoor air (OA). Compared to continuous exhaust, the average ventilation rate was higher in CZ 13, 15 and 16 (by 0.015 to 0.036 ACH) but lower in CZs 3 and 10 (by 0.02 and 0.7 ACH respectively). Although case 7 is not 62.2 compliant, the duct leakage during non-heating or cooling operation contributes significantly to the overall ventilation rate because there is about 50 cfm of balanced leakage – or up to 0.2 ACH for CZ15. In CZs with higher furnace blower air flows, the 7% OA operating mode leads to supply flow rates close to 62.2 requirements (e.g., 140 cfm in CZ15 compared to 150 cfm that would be required to meet 62.2 for 1/3 time operation). In addition, the supply air flow interacts with the other building leakage and envelope pressures such that the total ventilation rate is higher than for an exhaust fan of the same air flow.

The second system has an outdoor air flow rate set equal to the minimum 62.2 air flow rate that resulted in lower OA flows than for the 7% OA case. This resulted in yearly average effective ventilation rates that were less than continuous exhaust by 0.017 to 0.074 ACH depending on climate.

Table 5 Mean Annual ACH											
	Simulation										
Climate Zone	1 Unvented House *not 62.2 compliant*	2 Cont. Ex.	3 Int. Ex.	4 HRV 62.2 match air flow	4X HRV 50% ontime	5 CFI with Cont. Ex.	6 Supply	7 CFI 7%OA *not 62.2 compliant*	8 CFI 62.2 33% runtime *not 62.2 compliant*	Unvented House with 0.35 ACH Adder *not 62.2 compliant*	Cont. Ex. with 0.35 ACH Adder *not 62.2 compliant*
1	0.26	0.38		0.36	0.55						
2	0.23	0.37									
3	0.24	0.37	0.35			0.44	0.46	0.30	0.31		
4	0.24	0.38									
5	0.23	0.37									
6	0.19	0.35									
7	0.21	0.36								0.53	0.63
8	0.19	0.36									
9	0.21	0.37									
10	0.20	0.37				0.45	0.42	0.35	0.29		
11	0.28	0.42									
12	0.27	0.40									
13	0.23	0.38	0.35			0.47	0.44	0.39	0.33		
14	0.26	0.41									
15	0.24	0.42	0.39			0.51	0.45	0.45	0.35	0.52	0.59
16	0.32	0.43	0.41	0.42	0.61	0.53	0.55	0.47	0.41	0.54	0.56

Table 6 Effective Annual ACH											
	Simulation										
Climate Zone	1 Unvented House *not 62.2 compliant*	2 Cont. Ex.	3 Int. Ex.	4 HRV 62.2 match air flow	4X HRV 50% ontime	5 CFI with Cont. Ex.	6 Supply	7 CFI 7%OA *not 62.2 compliant*	8 CFI 62.2 33% runtime *not 62.2 compliant*	Unvented House with 0.35 ACH Adder *not 62.2 compliant*	Cont. Ex. with 0.35 ACH Adder *not 62.2 compliant*
1	0.24	0.37		0.34	0.54						
2	0.22	0.37									
3	0.22	0.36	0.34			0.43	0.45	0.29	0.30		
4	0.22	0.37									
5	0.22	0.36									
6	0.18	0.35									
7	0.20	0.35								0.52	0.63
8	0.18	0.35									
9	0.19	0.36									
10	0.19	0.36				0.44	0.40	0.34	0.28		
11	0.25	0.40									
12	0.24	0.38									
13	0.21	0.37	0.34			0.46	0.43	0.38	0.31		
14	0.24	0.39									
15	0.21	0.40	0.37			0.49	0.43	0.44	0.33	0.52	0.58
16	0.29	0.41	0.39	0.39	0.59	0.52	0.52	0.44	0.39	0.54	0.55

Table 7 Ventilation Efficacy[15]											
	Simulation										
Climate Zone	1 Unvented House *not 62.2 compliant*	2 Cont. Ex.	3 Int. Ex.	4 HRV 62.2 match air flow	4X HRV 50% ontime	5 CFI with Cont. Ex.	6 Supply	7 CFI 7%OA *not 62.2 compliant*	8 CFI 62.2 33% runtime *not 62.2 compliant*	Unvented House with 0.35 ACH Adder *not 62.2 compliant*	Cont. Ex. with 0.35 ACH Adder *not 62.2 compliant*
1	0.94	0.98		0.96	0.98						
2	0.94	0.98									
3	0.93	0.98	0.97			0.98	0.98	0.95	0.95		
4	0.94	0.98									
5	0.94	0.98									
6	0.95	0.98									
7	0.95	0.98								0.99	0.99
8	0.94	0.98									
9	0.94	0.98									
10	0.92	0.97				0.98	0.97	0.97	0.96		
11	0.90	0.96									
12	0.91	0.97									
13	0.92	0.97	0.98			0.98	0.97	0.97	0.96		
14	0.90	0.96									
15	0.90	0.95	0.96			0.97	0.96	0.97	0.94	0.99	0.98
16	0.89	0.96	0.96	0.93	0.96	0.97	0.95	0.95	0.94	0.99	0.98

[15] Ratio of Effective ACH to mean ACH.

Energy use

The energy use results are all expressed in Time Dependent Value kBtu using conversions from kWh and natural gas therms provided by the Commission. These calculations provide a different value to electricity energy use for each hour of the year for each climate zone. The results are all expressed in terms of site energy.

Case 1. Unvented House

In most CZs (except 15), natural gas for heating dominates energy use. The climates near the coast (CZs 1 through 7) and mountain climate (CZ 16) have very small cooling energy use (<5000 TDVkBtu) and CZ15 has twice the cooling energy use of the next CZ. The only significant effect of using TDV rather than energy use was that CZ13 used more energy in kWh but less TDV when compared to CZ14.

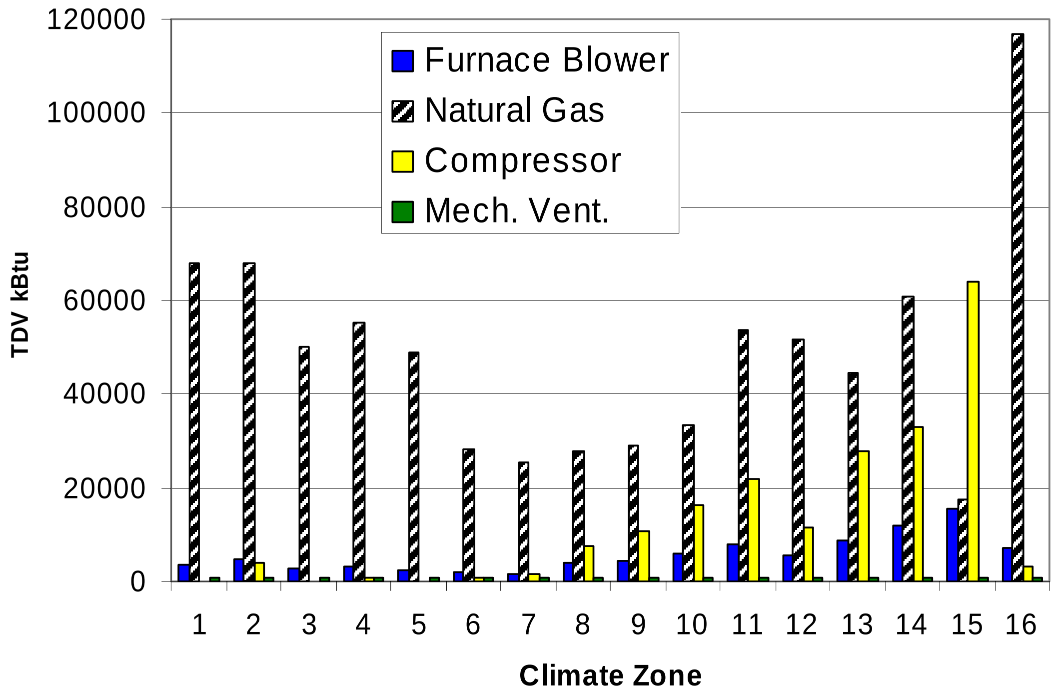

Figure 1. TDV Energy use in medium-sized unvented homes. Mechanical ventilation is intermittently-operated kitchen and bath fans only.

Continuous Exhaust vs. Unvented House (Case 2 vs. Case1)

Here we compare the 62.2 compliant continuous mechanically ventilated house to the naturally ventilated house. The mechanical ventilation exceeds the minimum required by 62.2 by 25 cfm to allow for periodic turning off of the system (whether as part of a controlled ventilation system or by occupant intervention). In the simulations, this system operated continuously. The average increase in TDV energy over all the CZs is about 10% (or about 6,500 kBtu), and is dominated by increased natural gas use for winter heating (except for CZ 15 where electricity use dominates). For this reason, the relative energy cost of adding mechanical ventilation is greatest in climates with the greatest heating requirements: CZ 16 and CZ 1. On a percentage basis, the impact is greater for CZs 6 and 7 with their low baseline heating requirements. CZs 6, 7 and 8 showed reductions in cooling energy use because the extra ventilation in these relatively mild cooling climates led to increased ventilation cooling. In the mild climates (6 through 9) the energy to run the mechanical ventilation fans is close to that used to condition the air. In other climates the energy used to condition the air dominates.

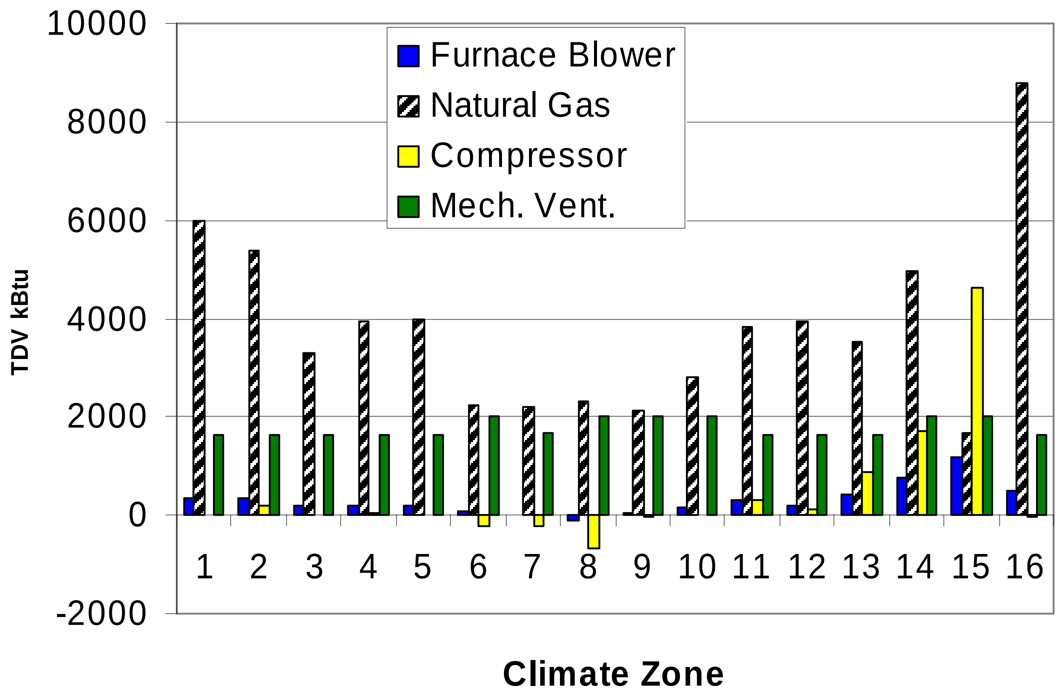

Figure 2. Change in TDV energy use for Continuous Exhaust compared to an Unvented House

Intermittent Exhaust vs. Continuous Exhaust (Case 3 vs. Case 2)

The average change in TDV energy due to the 17% reduction in bathroom fan operating hours is a decrease of about 2,500 kBtu or about 2.5%, and is mostly due to a combination of reduced cooling in CZs 13 and 15, reduced heating in 16, and a 13% reduction in mechanical ventilation fan power use in all 5 CZs examined.

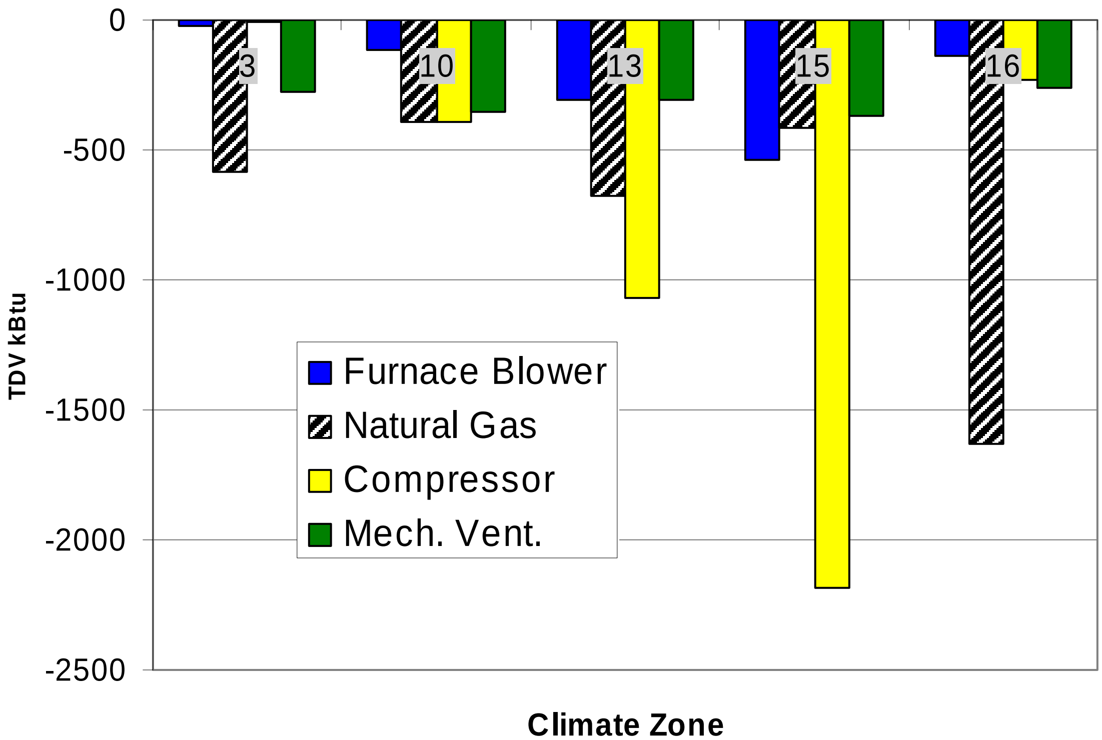

Figure 3. Change in energy use for Intermittent Exhaust compared to Continuous Exhaust reference.

HRV matching 62.2 airflow vs. Continuous Exhaust (Case 4 vs. Case 2)

The average change in TDV energy is a decrease of about 6,000 kBtu or about 5%, and is dominated by a reduction in the heating load and the consequent reduction in natural gas use. Because the HRV only operates for 10 minutes out of every hour, the mechanical ventilation power requirements are almost identical to those for continuous exhaust.

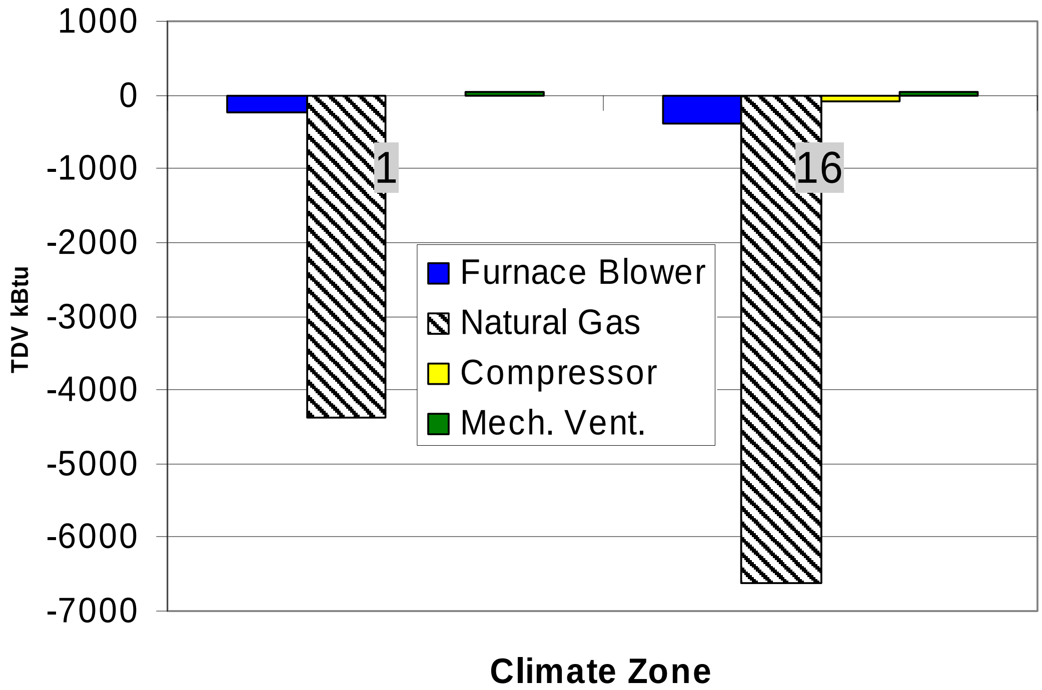

Figure 4. Change in energy use for HRV at 62.2 airflow compared to continuous exhaust reference

HRV at 50% ontime vs. Continuous Exhaust (Case 4X vs. Case 2)

The average change in TDV energy is an increase of about 1,000 kBtu or about 1%, and there is a balance between extra power requirements of the HRV fan and the reduction in heating requirements. It should be possible to increase the energy savings of the HRV if it were only operated in winter when the natural gas savings are realized. For the rest of the year, there are no savings to offset the HRV fan power use.

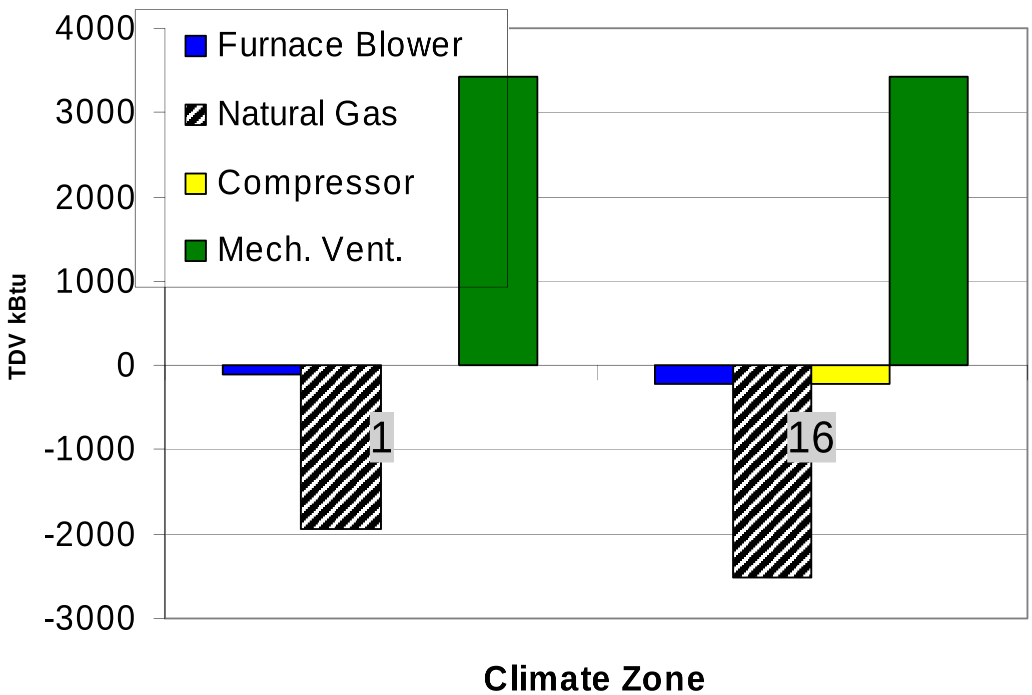

Figure 5. Change in energy use for HRV at 50% ontime compared to continuous exhaust reference

CFI with continuous exhaust vs. Continuous Exhaust (Case 5 vs. Case 2)

The average change in TDV energy is an increase of about 19,000 kBtu or about 22%, and is dominated by power requirements of the furnace blower that is used to distribute ventilation air. The furnace blower energy use could be reduced by using ducts, filters and coils with lower air flow resistance and utilizing electric motors that offer increased efficiency at the lower resulting external static pressures.

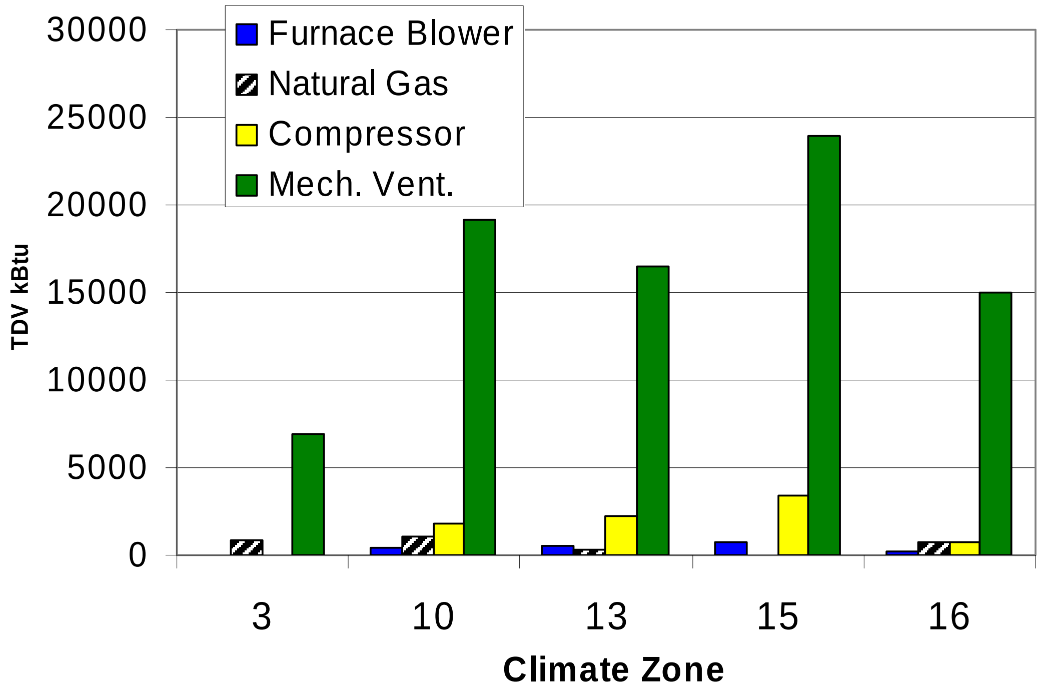

Figure 6. Change in energy use for CFI with continuous exhaust compared to continuous exhaust reference

Continuous Supply vs. Continuous Exhaust (Case 6 vs. Case 2)

The average change in TDV energy is an increase of about 11,000 TDVkBtu or about 13%, and is dominated by power requirements of the supply fan that has to move four times as much air as continuous exhaust to allow for tempering of outside air.

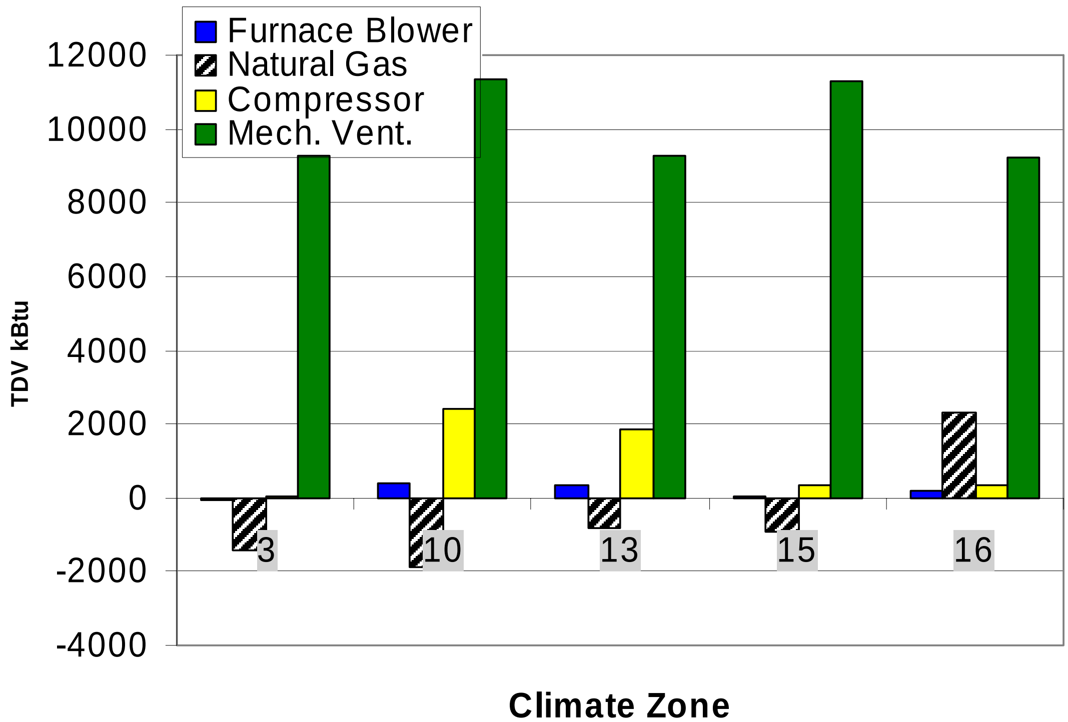

Figure 7. Change in energy use for continuous supply compared to continuous exhaust reference

CFI with 7% Outside Air (OA) vs continuous exhaust – not 62.2 compliant (Case 7 vs. Case 2)

The average change in TDV energy is an increase of about 16,000 kBtu or about 18%, and is dominated by furnace blower operation for ventilation.

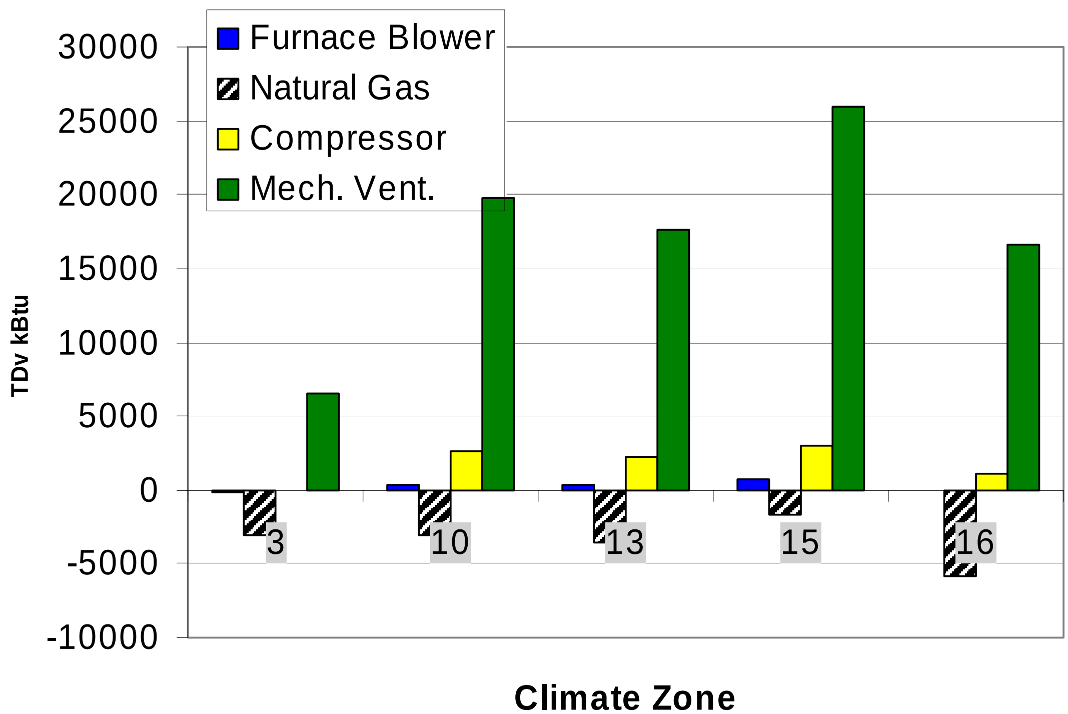

Figure 8. Change in energy use for CFI with 7% OA compared to continuous exhaust reference

CFI with 1/3 of 62.2 flow vs. continuous exhaust – not 62.2 compliant (Case 8 vs. Case 2)

The average change in TDV energy is an increase of about 17000 kBtu or about 20% and is completely dominated by furnace blower operation for ventilation.

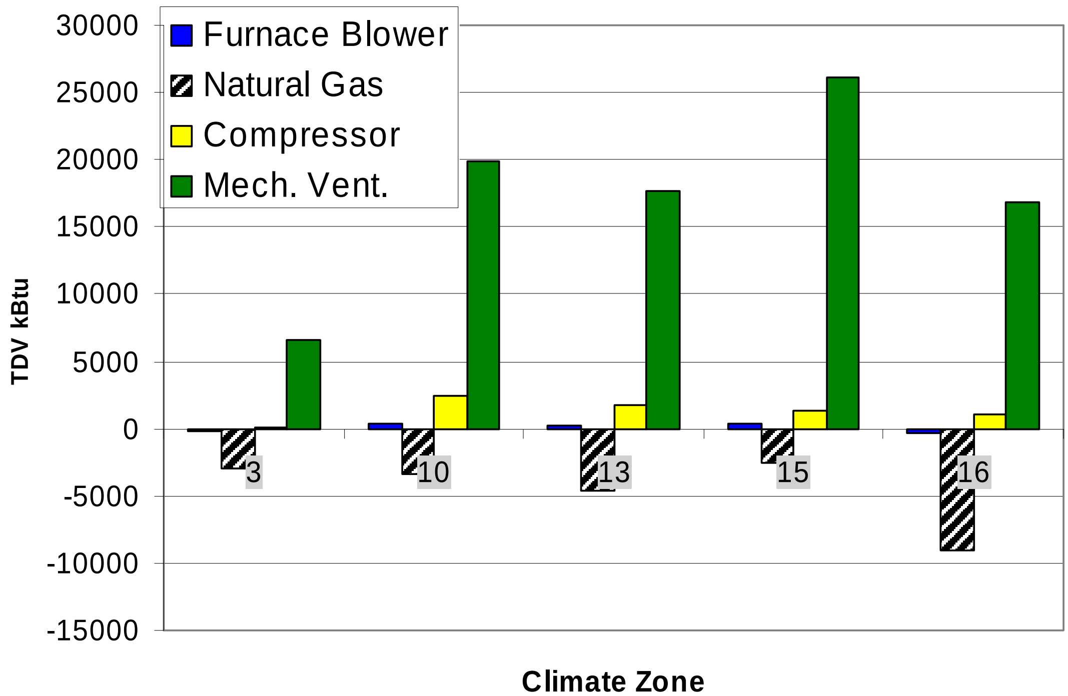

Figure 9. Change in energy use for CFI with 1/3 of 62.2 flow compared to continuous exhaust reference

Low-Ventilation Rate Adder (Case 9 vs. Case 2)

For the unvented house, the addition of 0.35 ACH when the ventilation rate falls below 0.35 ACH makes a significant difference. It adds 0.32 ACH to the mean ventilation rates in CZ 7 and 0.23 ACH in CZ 16 and adds 8 to 15% to HVAC energy use. In CZ 7 and CZ 16, the low ventilation rate adder reduces air conditioning use by 20% and 12% because these CZs have cool nights during the cooling season and the added 0.35 increases ventilation cooling. In CZ15, the consistently high outdoor temperatures preclude any ventilation cooling and 6% more cooling is required. For heating, 11% to 20% more natural gas is used depending on the climate with more energy used in colder climates.

For continuous exhaust, the effect of the added ventilation is 0.28 ACH for CZ 7 and 0.13 ACH in CZ 16. This is less than for the unvented house because the added ventilation is invoked less often. Overall the low-ventilation rate adder increased continuous exhaust energy use by 3% to 11%.

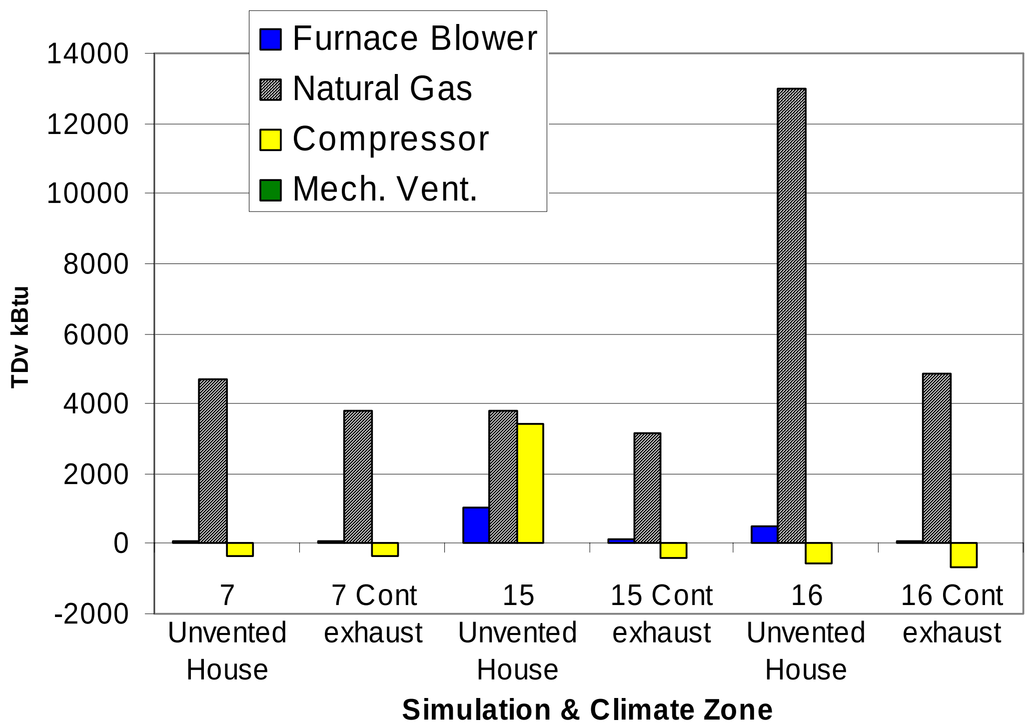

Figure 10. Effects of ventilation adder for unvented house and continuous exhaust

Peak Days Extra Ventilation Effects

A key question is how much does the extra ventilation contribute to power consumption on TDV peak days. Table 8 summarizes the energy use and average ventilation for the TDV electricity peak days in climates where there was more than 12 kWh of energy use for the TDV electricity peak day. These results are for the medium sized house with 62.2 compliant systems (except for the unvented house that is included as the base case).

Because these results are for a single day, they are not necessarily representative of overall system performance, but they do give useful indications of the effects of ventilation systems on peak TDV electricity cost days. These results show that the continuous exhaust systems add substantially to the ventilation rate on the peak day (by about 0.2 ACH), but the energy penalty is smaller than the change in ventilation rate, ranging from a reduction in energy use of 8% in CZ 8 to an increase of 8% in CZ 14. The intermittent exhaust in CZs 13 and 15 reduced ventilation at the peak time and reduced energy use by 5% and uses less energy than the unvented house in CZ13. The greater fan power requirements of the continuous supply system lead to energy use increases of 12% to 26% in CZs 10, 13 and 15. Lastly, the CFI systems that use the furnace blower use the most energy, with increases from 19% to 40%.

Table 8. Results from TDV peak days for Energy Use

CZ	Ventilation System	Average Indoor-Outdoor temperature difference C	Energ y kWh	Average ACH	Energy relative to unvented house, kWh	ACH relative to unvented house	Energy relative to unvented house %	ACH relative to unvented house %
8	Unvented House	0.4	13.1	0.16	-	-	-	-
	Cont. Ex.	0.2	12.0	0.37	-1.1	0.21	-8	126
9	Unvented House	-2.4	19.8	0.20	-	-	-	-
	Cont. Ex.	-2.5	19.4	0.41	-0.4	0.22	-2	109
10	Unvented House	-3.0	20.0	0.20	-	-	-	-
	Cont. Ex.	-3.2	21.7	0.42	1.7	0.21	8	105
	CFI + Cont. Ex.	-3.1	28.0	0.48	8.0	0.28	40	137
	Cont. Sup.	-2.9	25.1	0.38	5.1	0.18	26	87
12	Unvented House	-2.4	19.8	0.21	-	-	-	-
	Cont. Ex.	-2.7	20.3	0.42	0.5	0.21	3	99
13	Unvented House	-4.4	28.1	0.20	-	-	-	-
	Cont. Ex.	-4.3	29.3	0.43	1.2	0.23	4	54
	Int. Ex.	-4.5	27.8	0.39	-0.3	0.19	-1	44
	CFI + Cont. Ex.	-4.6	33.5	0.49	5.4	0.29	19	68
	Cont. Sup.	-4.2	32.3	0.38	4.2	0.18	15	42
14	Unvented House	-4.5	27.5	0.20	-	-	-	-
	Cont. Ex.	-4.5	29.7	0.43	2.2	0.23	8	82
15	Unvented House	-7.9	43.8	0.26	-	-	-	-
	Cont. Ex.	-7.9	46.8	0.49	3.0	0.24	7	55
	Int. Ex.	-7.9	44.8	0.45	1.0	0.19	2	45
	CFI + Cont. Ex.	-8.0	53.0	0.55	9.1	0.29	21	68
	Cont. Sup.	-8.0	48.9	0.41	5.1	0.15	12	36

Table 9. Results from TDV peak days in TDV$								
CZ	Ventilation System	Average Indoor-Outdoor temperature difference C	TDV$	Average ACH	TDV$ relative to unvented house	ACH relative to unvented house	TDV change %	ACH change as %
8	Unvented House	0.4	166.72	0.16				
	Cont. Ex.	0.2	146.27	0.37	-20.45	0.21	-12	126
9	Unvented House	-2.4	216.37	0.20				
	Cont. Ex.	-2.5	205.94	0.41	-10.43	0.22	-5	109
10	Unvented House	-3.0	150.28	0.20				
	Cont. Ex.	-3.2	152.77	0.42	2.49	0.21	2	105
	CFI + Cont. Ex.	-3.1	177.57	0.48	27.30	0.28	18	137
	Cont. Sup.	-2.9	186.13	0.38	35.85	0.18	24	87
12	Unvented House	-2.4	205.70	0.21				
	Cont. Ex.	-2.7	204.71	0.42	-0.99	0.21	0	99
13	Unvented House	-4.4	135.47	0.20				
	Cont. Ex.	-4.3	140.83	0.43	5.35	0.23	4	54
	Int. Ex.	-4.5	133.96	0.39	-1.51	0.19	-1	44
	CFI + Cont. Ex.	-4.6	149.91	0.49	14.44	0.29	11	68
	Cont. Sup.	-4.2	149.33	0.38	13.86	0.18	10	42
14	Unvented House	-4.5	213.69	0.20				
	Cont. Ex.	-4.5	229.98	0.43	16.29	0.23	8	82
15	Unvented House	-7.9	412.56	0.26				
	Cont. Ex.	-7.9	426.07	0.49	13.51	0.24	3	55
	Int. Ex.	-7.9	417.10	0.45	4.54	0.19	1	45
	CFI + Cont. Ex.	-8.0	456.51	0.55	43.95	0.29	11	68
	Cont. Sup.	-8.0	435.05	0.41	22.49	0.15	5	36

Table 9 shows the same results as in Table 8, but with TDV$ instead of energy. The TDV$ use the TDV hourly energy weightings together with a conversion from energy to dollars to give a number of TDV$ for this peak day. In general, the fractional (percentage) differences are smaller in TDV$ terms than energy terms. This implies that much of the energy differences must be at off-peak conditions. An illustration of this is shown in Figure 11 that compares the energy use, TDV$ and hourly TDV$ rate for the unvented house and CFI + continuous exhaust in CZ15 (this showed a big difference in percentage changes between Tables 8 and 9).

Figure 11 shows that when the TDV$ multiplier is high, the differences in energy use are small and the differences in energy use between the two cases are predominantly at off peak conditions when the TDV$ multiplier is relatively low. The small differences at peak are because the air conditioner is operating continuously at peak conditions no matter how the house is ventilated. The only difference in energy use on peak is the electricity used to power the ventilation systems that are independent of the furnace blower. The power consumption of these fans is insignificant compared to the power consumption of the air conditioner. In addition, the reduction in thermostat setpoint between hours 14 and 18 also tends to make the air conditioner operate continuously. This time period is coincident with the TDV$ peak. This combination of operating characteristics leads to the counter-intuitive result that applying peak TDV multipliers to electricity results in lowering the differences between ventilation systems.

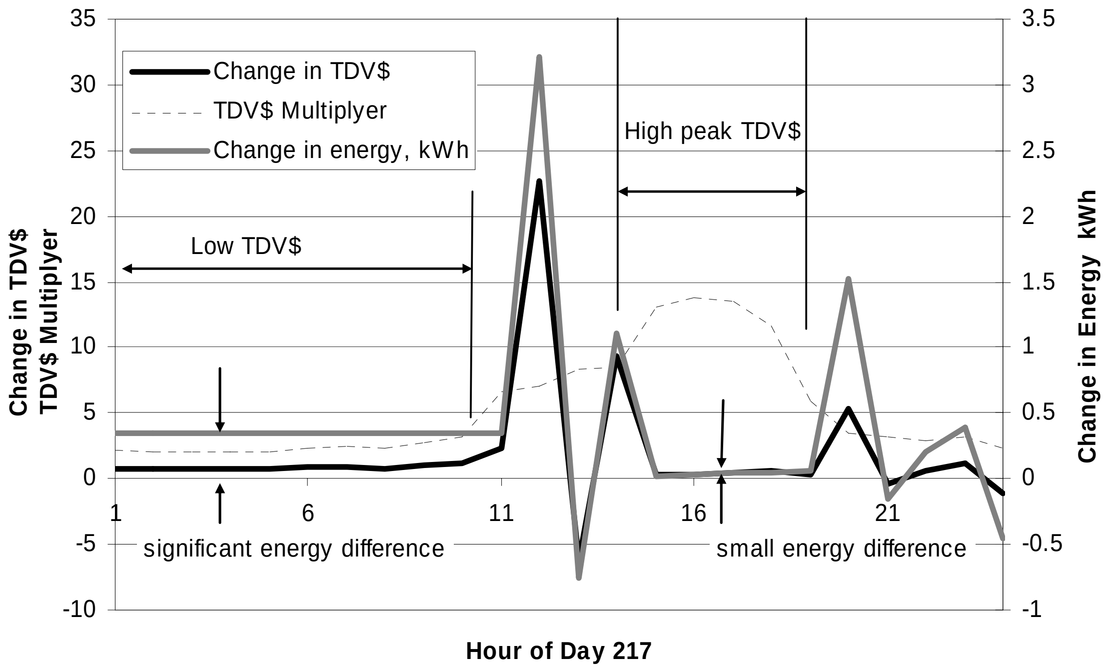

Figure 11. Comparison of time of energy use for CZ 15 between the unvented house and a CFI + continuous exhaust.

Indoor Concentrations of Pollutants at Low Ventilation Rates

For indoor air quality issues, a key area of concern is at times of low ventilation, which is when indoor concentrations were at their highest. To compare the different systems, hours when the unvented house had low ventilation rates were selected (hours where the mean ACH was less then 0.1). The concentrations were calculated using the minute-by-minute air flows and assuming a constant indoor emission rate. The results were normalized by comparing to the indoor concentration that would occur for the same house constantly ventilated to the ASHRAE 62.2 rate of:

$$0.03 \times \text{floor Area (ft2)} + 7.5 \text{ cfm/person.}$$

This works out to be about 0.3 ACH.

The results (summarized in Appendix D) show that the unvented house often has indoor concentrations two to three times higher than the mechanically ventilated cases that meet ASHRAE 62.2.

Figure 12 shows the difference between a simple continuous exhaust system and the unvented house for each hour of the year where the unvented house had less than 0.1 ACH. The continuous exhaust results are much more uniform than the unvented house and are generally two to three times lower.

Figure 13 is more complex and includes the many systems examined in CZ10. However, there is still a clear delineation between the unvented house and mechanically ventilated houses, even for those systems that do not meet 62.2 (CFI 7% OA and CFI 1/3 62.2).

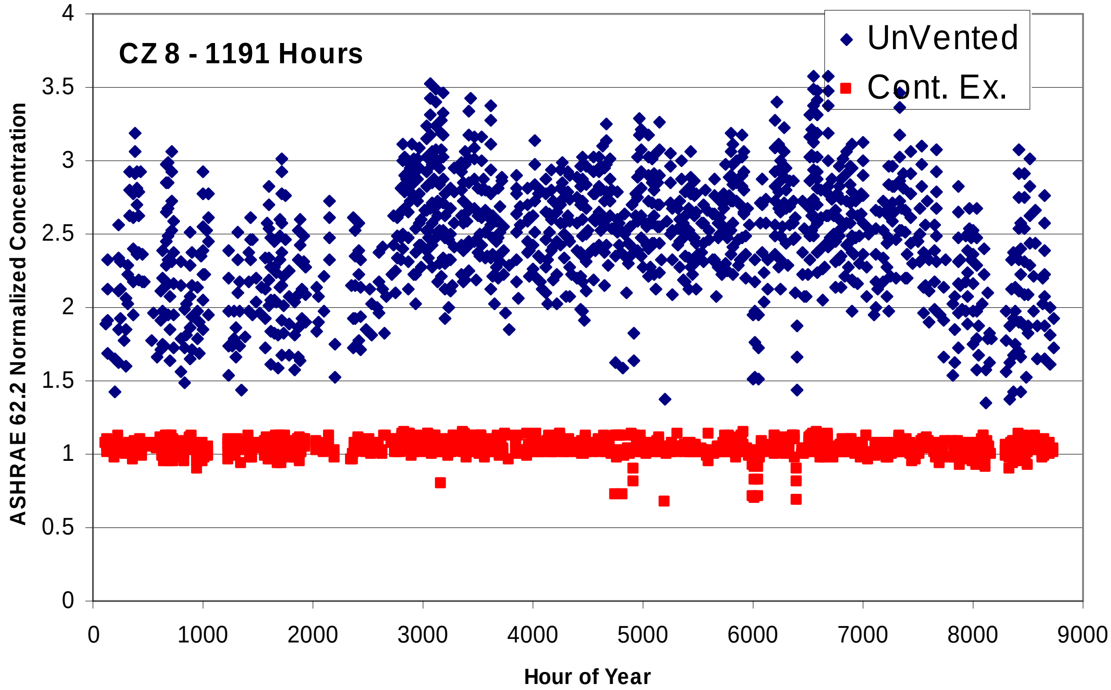

Figure 12. Climate zone 8 indoor air concentrations for periods of low air change rate in an unvented house

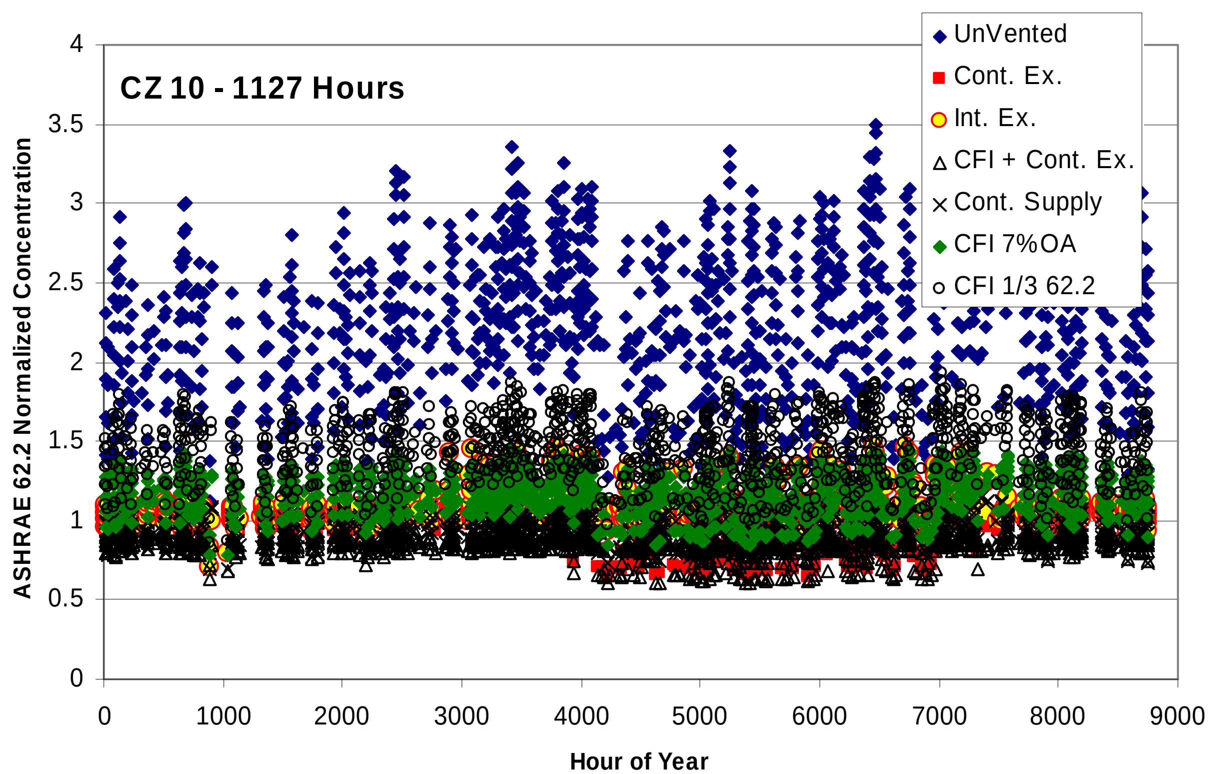

Figure 13. Comparison of indoor concentrations for CZ 10 at low air change rates

Delivered Air Temperatures for Supply Ventilation

Delivered air temperatures are important for supply air systems because if air is supplied at too high or low a temperature, then occupant comfort will be compromised. For the HRV, continuous supply, and the CFI with continuous exhaust, we determined the supply air temperatures for every minute of the year. The data were then binned by temperature so that we can see how often a particular delivered air temperature occurs during the year.

For the HRV, the delivered air temperature (T_{del}) is determined from the apparent sensible effectiveness (0.7) and from the indoor (T_{in}) and outdoor (T_{out}) temperatures:

$$T_{del} = 0.7T_{in} + 0.3T_{out}$$

The HRV was operated in two modes – either 10 minutes out of each hour (10/60) or half of each hour (30/60).

Similarly, the continuous supply mixes indoor air with outdoor air in a ratio of 3:1 such that:

$$T_{del} = 0.75T_{in} + 0.25T_{out}$$

Lastly, the CFI system mixes outdoor air with circulating air at about a ratio of 1:15. There is also heat transfer in and out of the duct system that can change the delivered air temperature. The CFI temperatures were split into five categories:

1. CFI with no heating or cooling
2. CFI when heat is also on
3. CFI when cooling is also on
4. Heating only – CFI duct closed
5. Cooling only – CFI duct closed

These 5 categories allow us to see how much CFI operation changes delivered air temperatures when the system is heating and cooling as well as when the CFI is operating for ventilation only (Category 1).

For climate zone 1, only the HRV was simulated. The results in Figure 14 show that for a few minutes of the year (210 minutes for 10/60 operation and 660 minutes for 30/60) the delivery temperatures get as low as 11°C (52°F) but the majority of operation is in the 15/16°C range (59-61°F). This suggests that supply vents should be carefully placed so as not to blow directly on occupants.

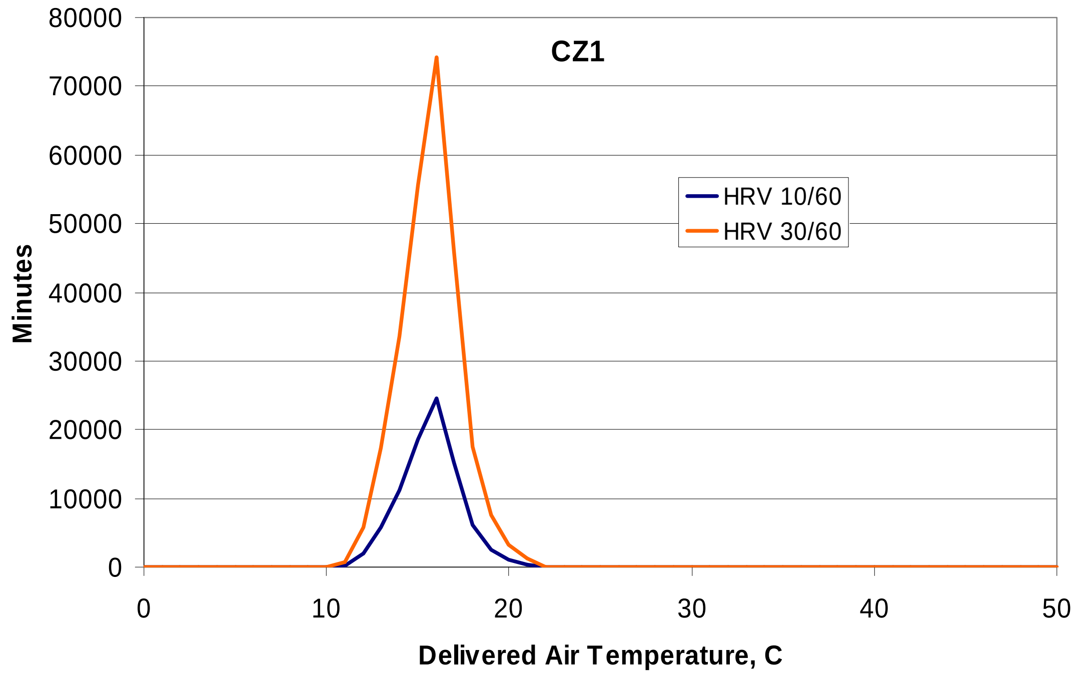

Figure 14. Distribution of delivered air temperatures for HRV's in CZ1.

For Climate Zone 3, we simulated the CFI and continuous supply. Figure 15 shows that, when there is no heating or cooling, the CFI delivers air that is at less extreme temperatures than the continuous supply because it mixes outdoor air with more indoor air. Both systems have supply air temperatures low enough such that care must taken to avoid cold drafts for occupants. Another possibility would be to use controls that turn off the system when outdoor temperatures are low (e.g., below 0°C (32 F)) because at these low ambient temperatures the natural infiltration through the envelope is high so that turning off the mechanical ventilation does not result in ventilation rates that are too low. Heating with the CFI duct open results in a wider range of delivered air temperatures compared to heating when the CFI duct is closed. This shifts the median delivered temperature from 48°C (118°F) to 46°C (115°F) but never delivering air below 36°C (97°F). The CFI operation tends to spread out the delivered air temperatures when operating in conjunction with heating and cooling. In CZ 3 there are a few minutes (only 424 minutes (about 7 hours) for the whole year) of cooling but not enough to be directly visible in the figure.

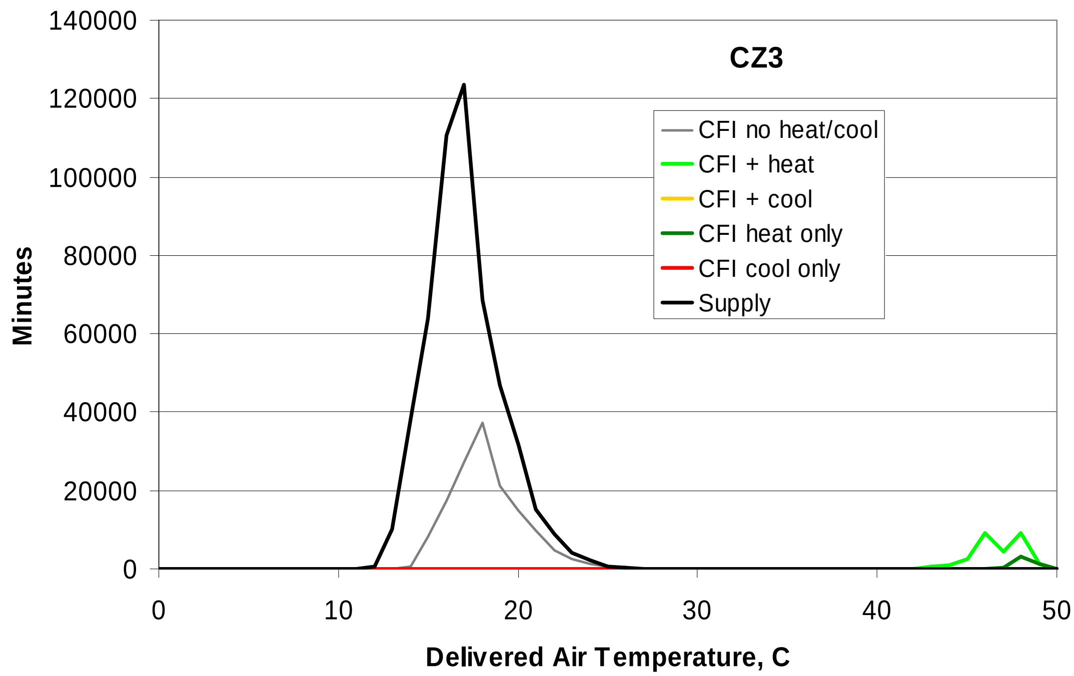

Figure 15. Distribution of delivered air temperatures for CZ3

For Climate Zone 10, 13 and 15 (Figures 16 through 19), we simulated the same systems as CZ 3. In general, the results are similar with the CFI consistently delivering air in a narrower range of temperatures than continuous supply. This is most apparent in CZ16 where the CFI does not supply air below 14°C (57°F), but the continuous supply goes down to 9°C (43°F). CZ16 also includes HRV's, and as in CZ3 they tend to produce the lowest delivered air temperatures. Note that the HRV's (and continuous supplies) in these simulations were not linked to heating or cooling system operation and they could be installed (together with the appropriate controls) to synchronize heating and cooling with ventilation to provide tempering of extreme delivery temperatures.

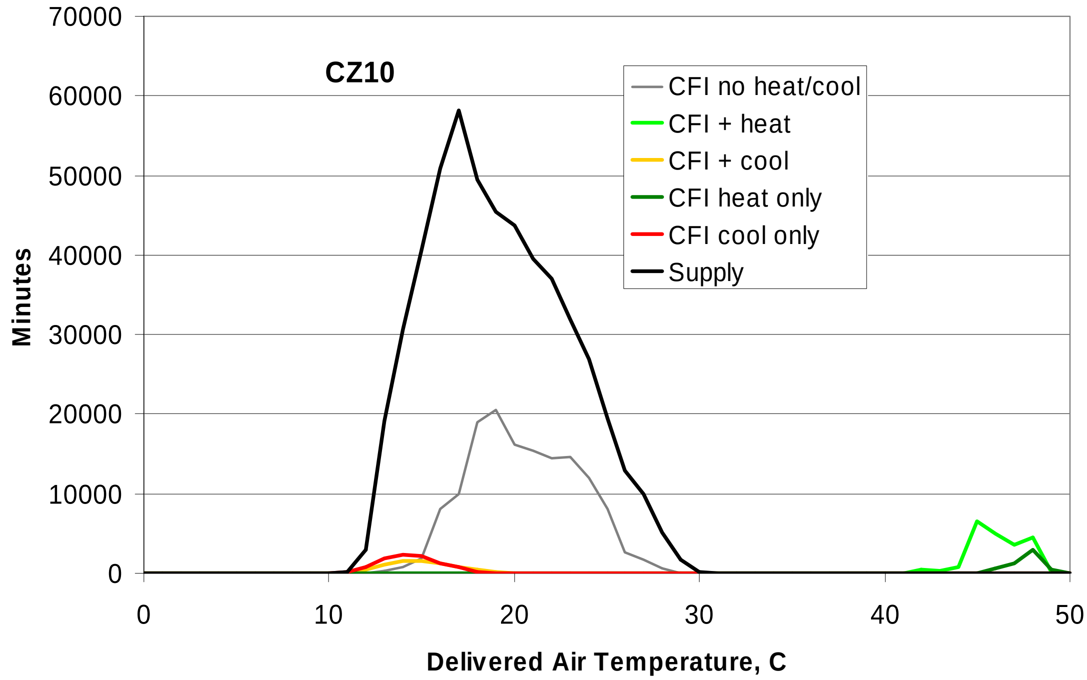

Figure 16. Distribution of delivered air temperatures for CZ10.

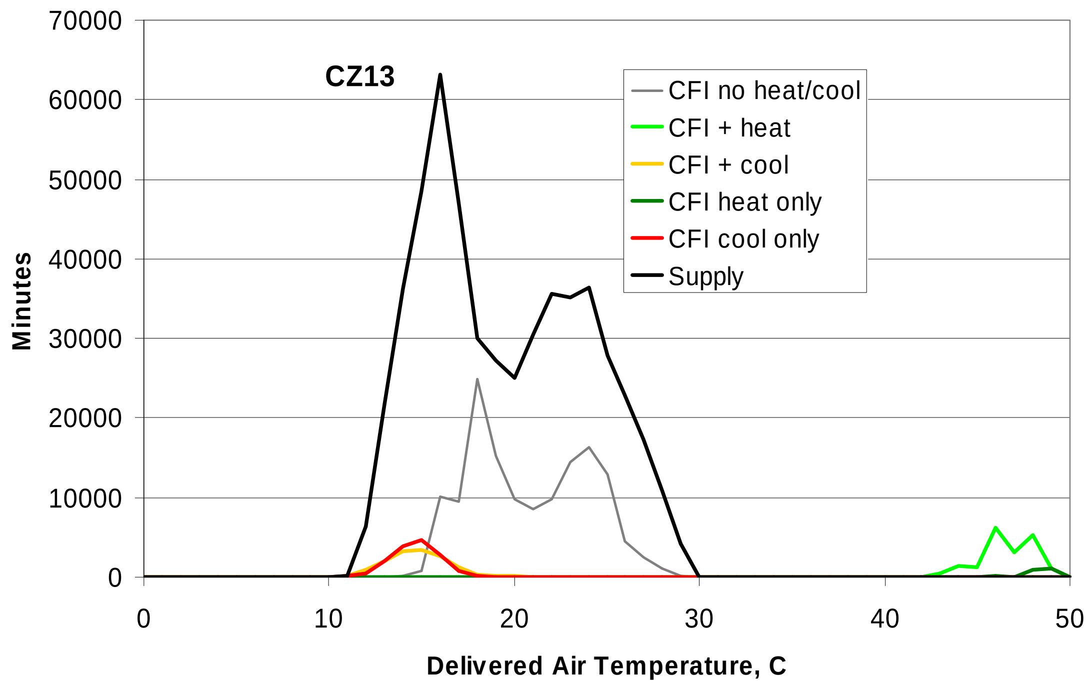

Figure 17. Distribution of delivered air temperatures for CZ13.

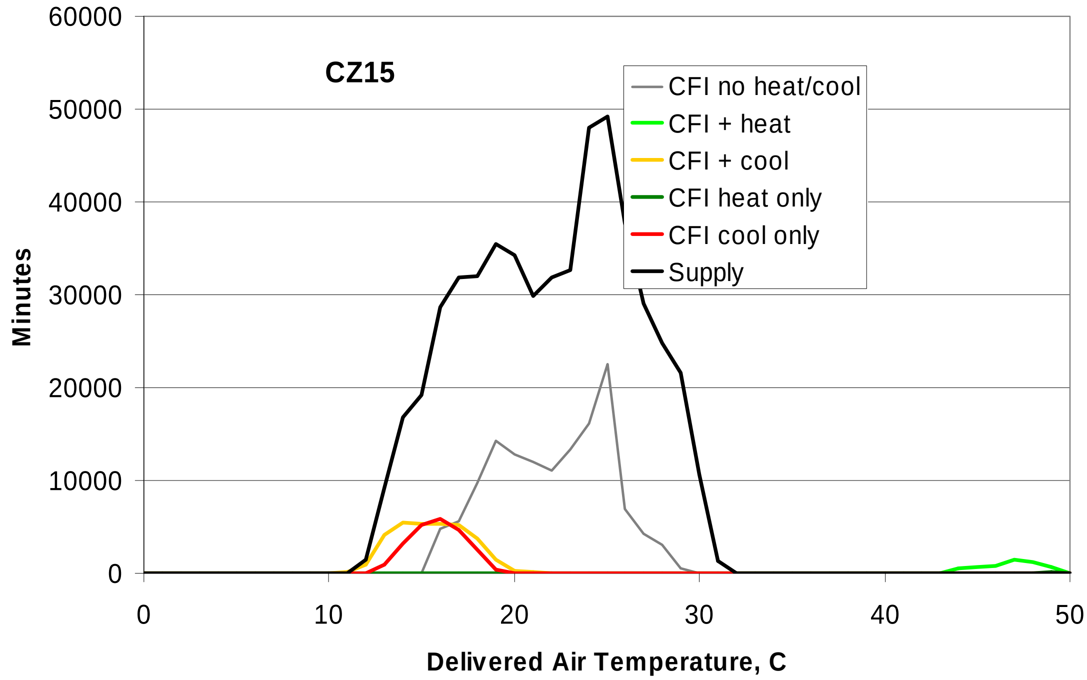

Figure 18. Distribution of delivered air temperatures for CZ15.

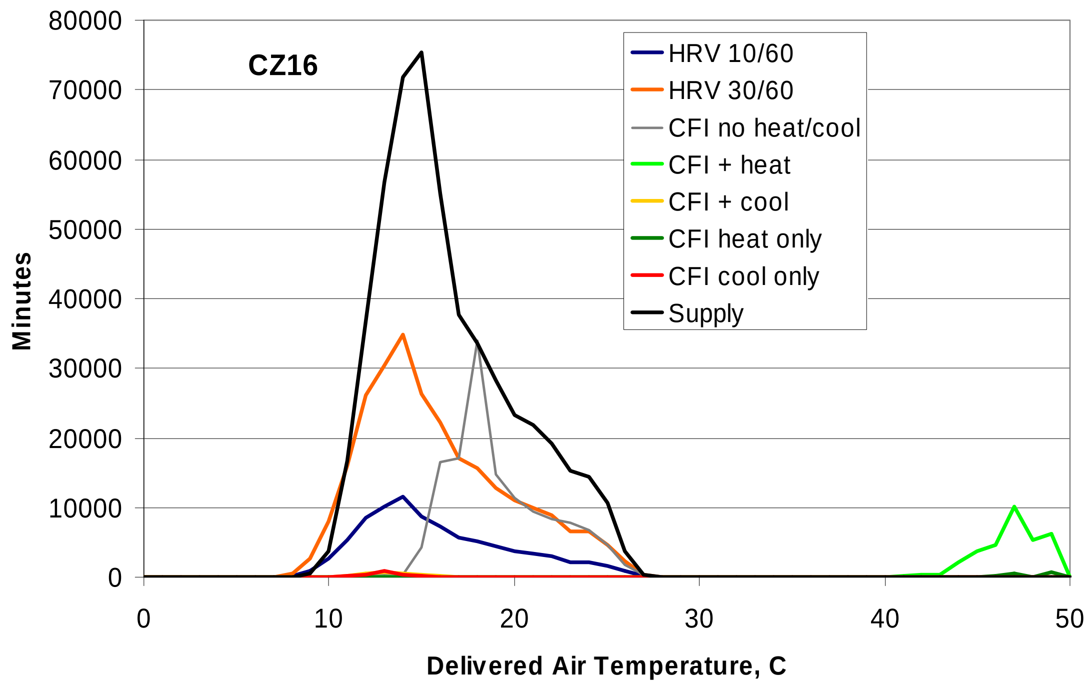

Figure 19. Distribution of delivered air temperatures for CZ16.

Effects of Leaky Ducts on CFI performance

To examine the effect of leaky ducts on CFI performance, simulations were performed for CZs 3, 13 and 16 with duct leakage increased from 5% to 11%. The results in Figure 20 show that the TDV increases significantly (9% on average for these three climates), indicating that the CFI system should only be used with tight ducts. Because more time is spent heating and cooling (leading to increases in heating and cooling energy), there is a slight decrease in the TDV attributed to mechanical ventilation because the CFI operates for less time without heating or cooling.

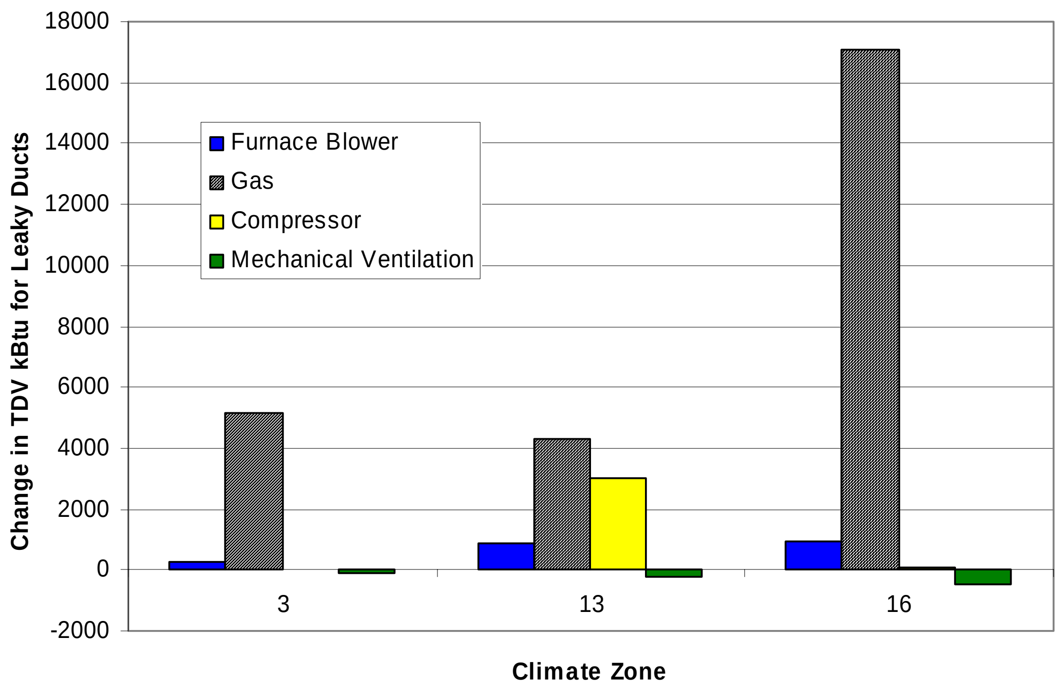

Figure 20. Effect of increasing duct leakage from 5% to 22% on CFI system performance.

House size

Three different house sizes were simulated to examine the effect on overall energy use (used for peak demand and consumer cost) and energy use per square foot of floor area. House size effects were examined for continuous exhaust, intermittent exhaust, and CFI with continuous exhaust.

For the continuous exhaust case, CZs 3, 10, 13, 15 and 16 were simulated for all three house sizes. The L, M and S after each climate zone indicate Large, Medium or Small house. The results in Figure 21 show the variability with house size, and clearly demonstrate the large energy savings for smaller houses.

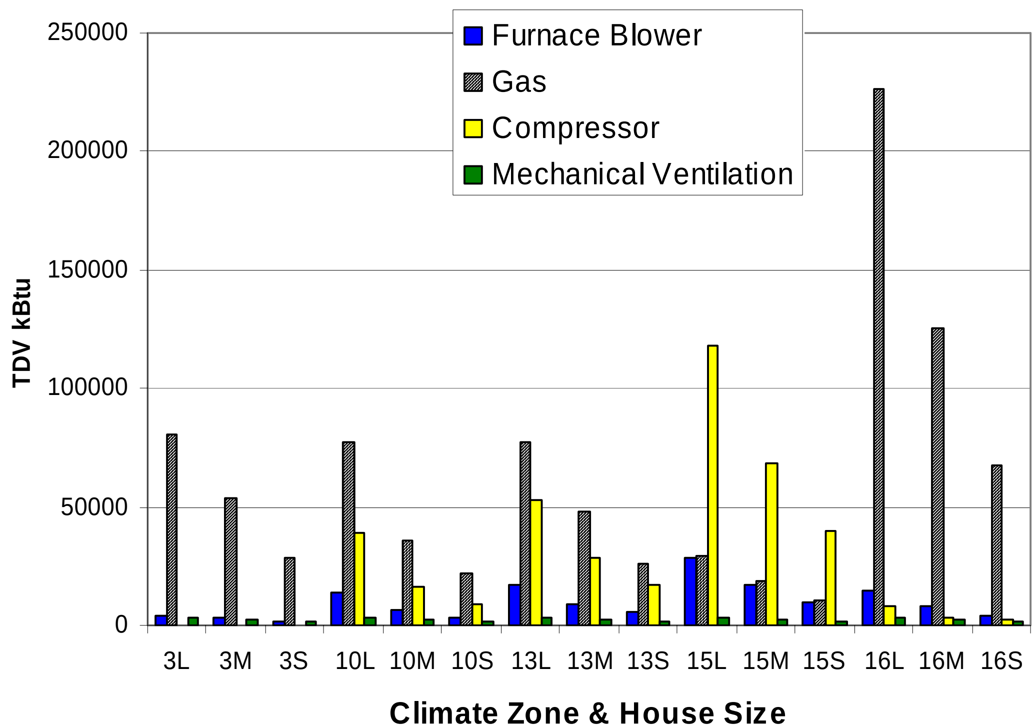

Figure 21. TDV energy use for continuous exhaust for different house sizes.

If we normalize the energy use by house floor area (this is the metric used in the California Building Energy Code), we get the results in Figure 22. The results show that the large house that uses the most energy has the lowest rating when normalized by floor area. The Medium sized house uses more energy per square foot in most cases. The exception is cooling for 15S and heating for 10S that are largest.

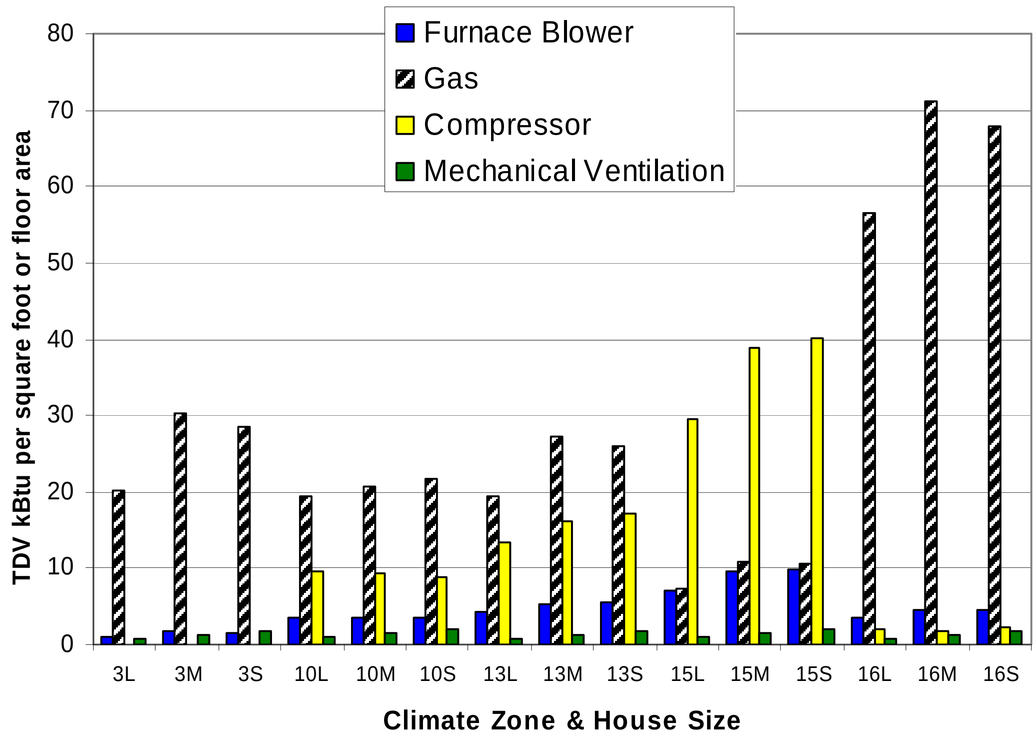

Figure 22. TDV Energy use for continuous exhaust normalized by floor area.

The air change rates in Figure 23 show that larger houses have less air changes per hour. This is expected because the sizing algorithm from ASHRAE 62.2 and the assumed occupancy do not scale directly with floor area.

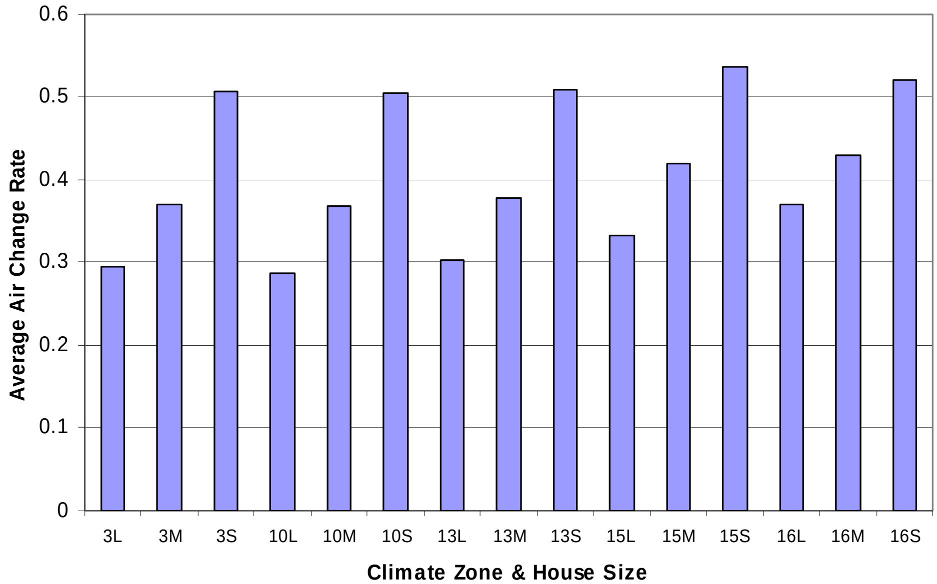

Figure 23. Average Air Change Rate for continuous exhaust for different house sizes.

The effect of house size with intermittent exhaust was investigated for climate zone 16 and is shown in Figure 24. This shows the expected scaling with house size. Figure 25 shows the same results normalized by floor area. As with the continuous exhaust case above, the medium house tends to have the highest energy use per square foot.

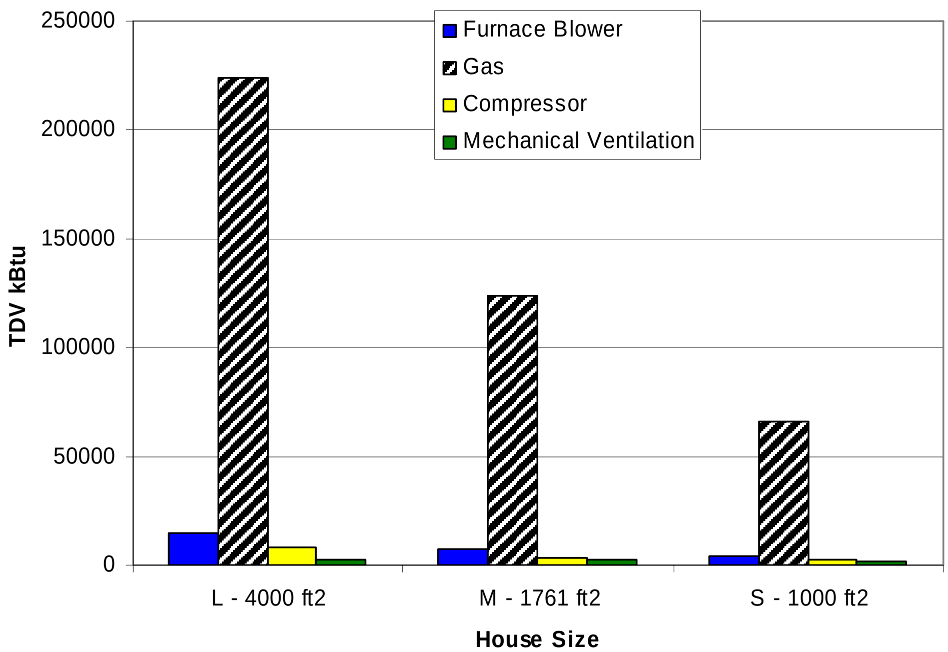

Figure 24. Effects of house size on TDV energy for intermittent exhaust in CZ 16

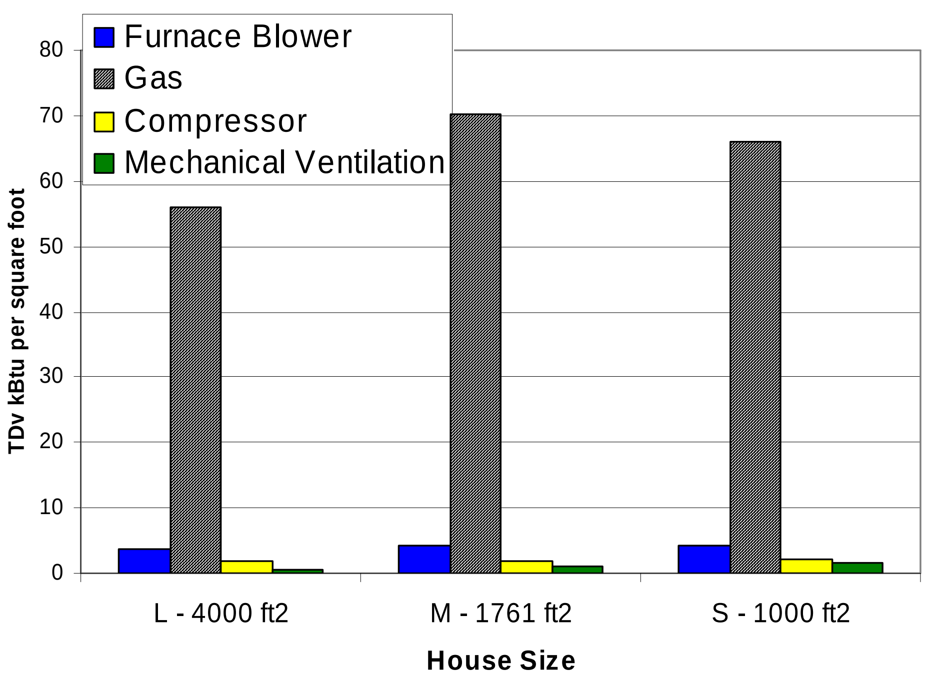

Figure 25. Effects of house size on TDV per square foot for intermittent exhaust in CZ 16.

The CFI (case 5) was examined in CZ 10 for three house sizes. This CZ is less dominated by heating (gas consumption) as shown in Figure 26. This shows how the large house consumes much more energy than the other houses. The results are also shown normalized by floor area in Figure 27. The differences are less marked than the comparisons above with little differences in the normalized data.

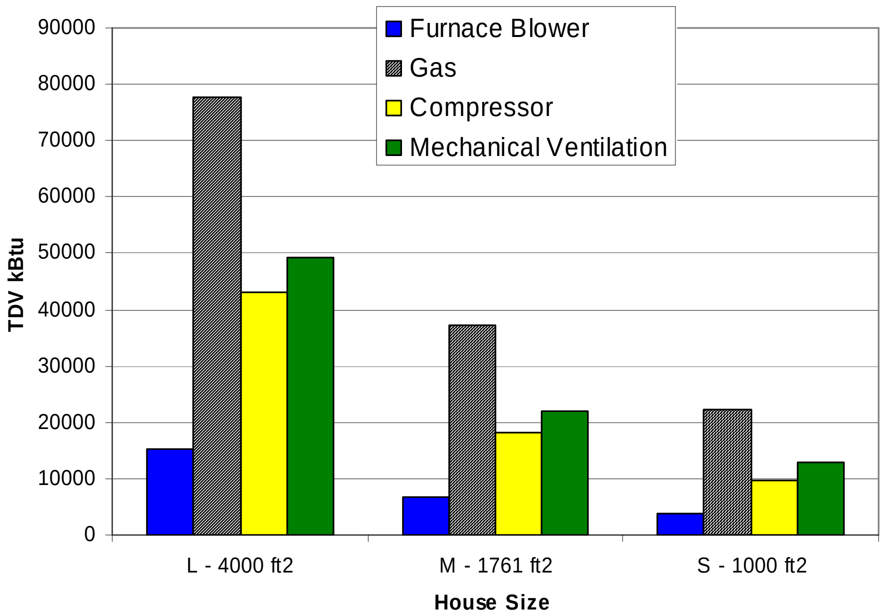

Figure 26. Effects of house size on TDV for CFI in CZ 10.

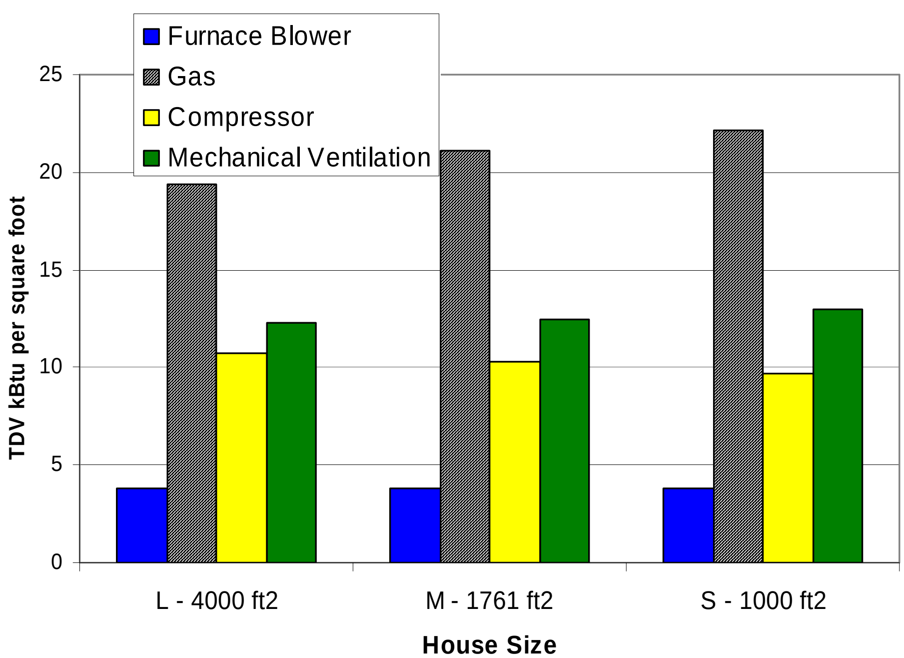

Figure 27. TDV Normalized by House size for CFI in CZ 10.

Effects of Intermittent Ventilation on Indoor Ozone

Usually the focus of ventilation is to provide fresh outdoor air and remove stale indoor air because the source of air pollutants is from inside the house. In some cases, however, the outdoor air may contain pollutants that we do not want to bring into the house. One important outdoor pollutant in California is ozone. When developing the list of acceptable technologies for ventilating California houses, we considered the ability of technologies to operate intermittently – thus giving the flexibility to ventilate less at peak load and also to ventilate less if there are undesirable outdoor pollutants. The simplest example of this is the use of an intermittent exhaust fan. Here we will compare the air flow rates and resulting transport of ozone into a house for both continuous and intermittent exhaust.

Data for hourly outdoor ozone concentrations were obtained from the California Air Resources Board website for Riverside, CA (in climate zone 10). The peak ozone concentration day is August 14th. Because the TMY weather data and the ozone concentration data are not taken at the same time (they are for completely different years), we chose to use ventilation rates taken from a typical day that was nearest to design temperature conditions. In this case, for CZ10, September 3rd was used. The corresponding hourly averaged ventilation rates were then used to calculate the amount of ozone entering the house each hour. We did not calculate indoor concentrations because they depend on many things that we are not modeling for this study: deposition in the envelope of the building, interaction with indoor surfaces, etc. Instead we can provide a relative measure of how much potential there is for reducing ventilation-related ozone entry.

Figure 28 illustrates the effects of the intermittent and continuous ventilation strategies on the quantity of ozone entering the house envelope by ventilation. The differences occur in the four afternoon/evening hours when the intermittent ventilation is off. Because the reduced ventilation is coincident with peak ozone concentration, the effect on the quantity of ozone delivered is significant. In this example, the reduction in total ozone for the day was almost 20%. For the four hours of reduced ventilation, the average difference was 40% and the greatest reduction for a single hour was 50%.

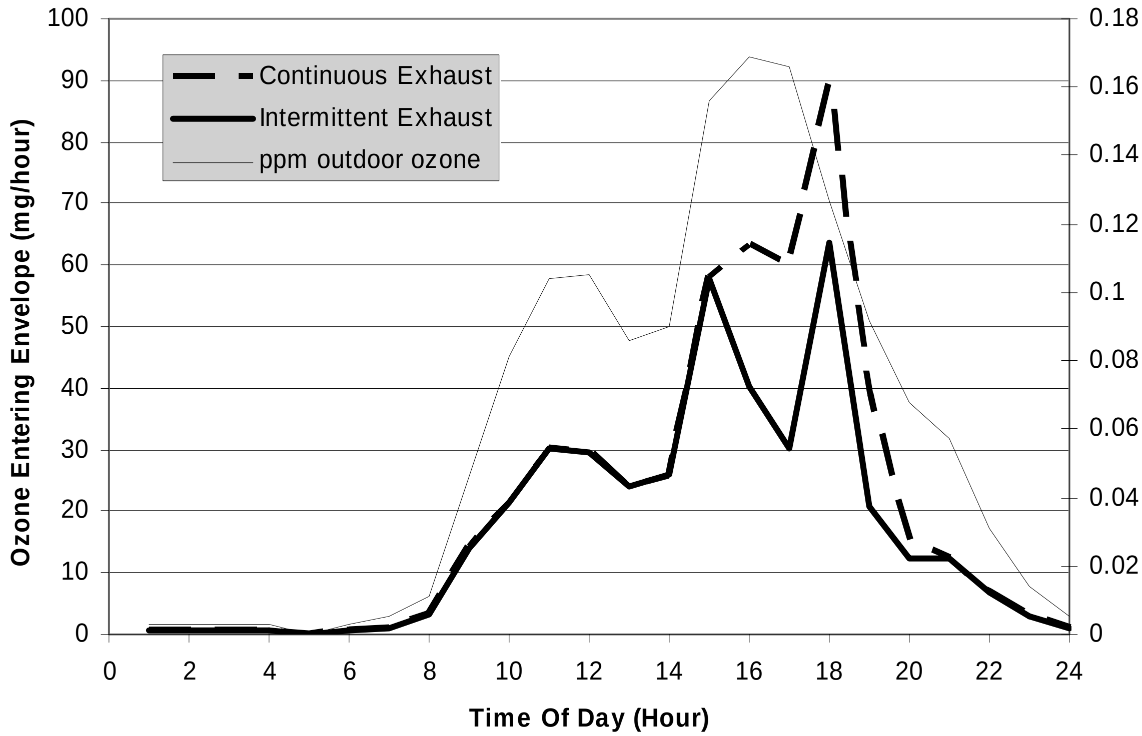

Figure 28. Outdoor concentration profile and ozone delivery rates for Riverside CA with continuous and intermittent exhaust.

Economizers and Intermittent Ventilation

The principles of intermittent ventilation can allow one to reduce ventilation during some periods and to ventilate at greater than the minimum the rest of the time. This ability is most useful when ventilation is substantially increased for some reason independent of supplying minimum ventilation, such as for ventilative cooling.

In many climates, the use of air exchange (i.e. flushing) to remove internally generated heat can be an energy savings strategy. Whether this is done by opening windows or though a mechanical system, such as an economizer, the impact on indoor air quality is the same: there is substantial more flushing of internally-generated contaminants.
Once the flushing is over, it is possible to delay any mechanical ventilation for a period, and still get equivalent exposures to that assumed by a constant ventilation rate. Figure 29 was generated using intermittent ventilation equations[16] applied to a situation in which a large amount of flushing (at least 10 times the rate in 62.2) was used for a number of hours.

The bold solid line indicates the number of hours of time that the mechanical ventilation system can be shut-off following a known length of flushing. For example, after 8 hours of flushing time (typical for nighttime economizer operation), the system can be shut off for about 11.5 hours. The dashed line is the peak concentration relative to the steady-state value that would result from that practice. Following the above example, after 8 hours of flushing and 11.5 hours with the system off, the peak concentration is about four times the steady-state value. If the contaminants of concern have non-linear dose-response curves (e.g. threshold values), this peak level could be important.

The thin solid line is the curve representing a single day (i.e. the number on the x-axis and the number on the y-axis add to 24). The crossing of the thin and bold lines mean that no additional mechanical ventilation is needed that day to meet minimum ventilation requirements. Thus, if an economizer is running for at least 12 hours a day, no other mechanical ventilation is required that day. This could save substantial energy (and peak power) in hot, dry climates where flushing is commonly done at night, but air conditioning is required during the day.

[16] From Sherman. 2006. Efficacy of Intermittent Ventilation for Providing Acceptable Indoor Air Quality, ASHRAE Trans. Vol 112, pt. 1. ASHRAE, Atlanta, GA.

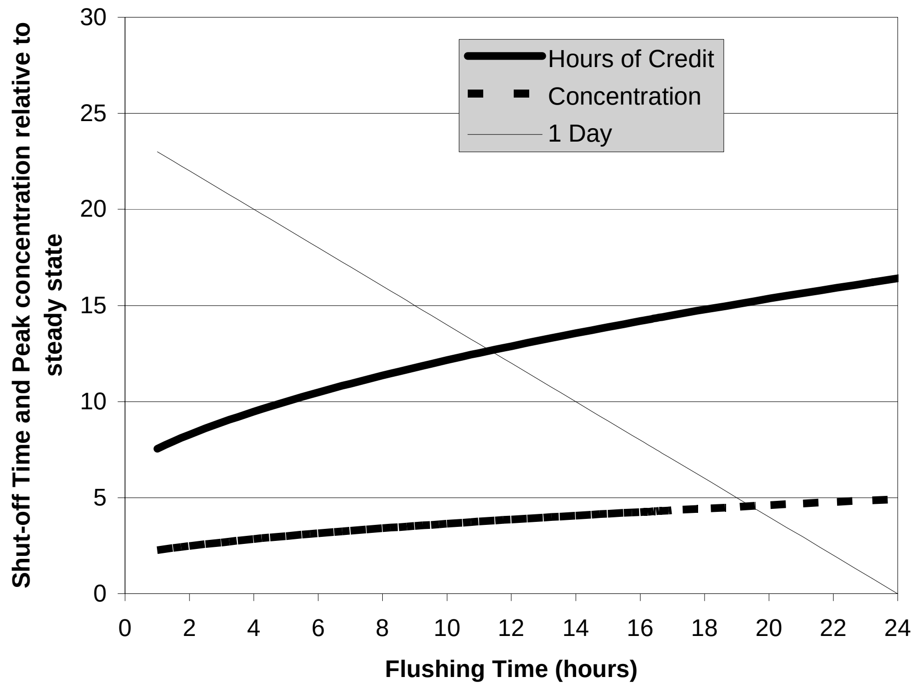

Figure 29. Hours of Credit and Peak Concentration Effects For High Air flow Flushing Ventilation

Summary

This study used simulations to examine the effect of different ventilation strategies on energy use in California houses. The houses were California Building Energy Code compliant, but not all the ventilation related energy use used in the Alternative Calculation Manual were implemented (e.g., maximized ventilation cooling). The simulations focused on ventilation technologies that were complaint with ASHRAE Standard 62.2. Extra simulations were performed for some systems that are commonly used in California for mechanical ventilation, but do not meet ASHRAE 62.2 minimum requirements.

The simulation results have shown that ASHRAE Standard 62.2 compliant ventilation systems add significantly to ventilation rates and reduce indoor pollutant concentrations; however, there is a cost associated with this extra ventilation. For a minimally compliant continuous exhaust system the extra TDV energy use is about 10%. For perspective, this is about the same TDV energy change as the 0.35 ACH ventilation adder used at low ventilation rates in the current California Building Energy Code Alternative Calculation Manual. Relative to the minimally ASHRAE 62.2 compliant exhaust fan, the intermittent exhaust and HRV systems reduced TDV energy use by 1% to 5%. The CFI and supply systems averaged 22% and 13% more energy than the continuous exhaust respectively.

In terms of TDV, the energy required for ventilation was dominated by natural gas use for heating in most climate zones (except CZ15 where there was more energy used for cooling). The ventilation fan power requirements for continuous exhaust fans were about half the extra space conditioning extra load on average. In mild climates (6 through 9) the fan energy was about the same as the conditioning energy, but in other climates the conditioning energy dominates.

HRV energy use was dominated by the energy used to operate the HRV fans. Because HRV’s give the greatest benefit at high temperature differences, operating all year when temperature differences are small allows the fan energy to offsets the space conditioning benefits. Also, the limited air flow range of available HRV’s meant that the airflows they provided significantly exceeded the 62.2 minimums. In this study we found that the HRV only needed to operate for 10 minutes out of each hour to have the same average ventilation rate as a minimally compliant 62.2 system. This would reduce the HRV fan energy requirements.

CFI systems also provide distribution and mixing of air. This is an extra service beyond the basic requirements of 62.2. In developing code requirements, the Commission needs to decide how to deal with this benefit. The 62.2 compliant CFI system that we studied used a continuously operating exhaust to meet 62.2 and the CFI provided extra ventilation when operating (due to the change to balanced ventilation from exhaust). It could be argued that the cost of operating the CFI should not be included in the ventilation estimates because it is providing another separate although complimentary service.

Two non-ASHRAE 62.2 compliant CFI systems were also investigated that are currently used in California construction. The amount of ventilation provided by these CFI systems depends on the size of opening to outside and the air flow through the duct system. Therefore, the amount of ventilation depends on system capacity for a given fraction of outside air. Lastly, the CFI systems all have additional ventilation from duct leakage when operating. To prevent excess ventilation, their duct systems need to be as tight as possible. Even the 5% total leakage used here (2.5% each for supply and return) leads to ventilation flows that are significant (typically half of the 62.2 minimum rate).

Intermittent ventilation systems and strategies can be used to significantly reduce the effects of outdoor air pollutants. Intermittent exhaust can reduce the ozone delivered to the house by 50% at peak outdoor ozone concentration.

Significant credit (in terms of reduced mechanical ventilation operation) can be given for large ventilation air flows. Typical nighttime economizer operation for 6 to 8 hours would allow for 10 to 12 hours of no mechanical ventilation requirements. This would allow for reduced mechanical ventilation (and associated air conditioning and fan power electricity consumption) through most of the day – including the afternoon electricity peak.

Recommended Technologies

All the 62.2 compliant technologies studied here are recommended for use in California with the following caveats:

- All selected ventilation fans should use as little energy as possible. The low sone requirement of 62.2 effectively biases selections toward high efficiency models already.

- Intermittent exhaust allows flexibility of operation, energy savings, and the ability to reduce the effects of outdoor pollutants, but must still be sized and operated to meet 62.2.

- HRV use could be optimized by using air flow rates (or time of use that provides the ASHRAE 62.2 air flow required each hour – in the simulation presented here this was 20 minutes operation per hour).

- Supply systems (either dedicated continuous supplies or intermittent CFI systems) move a lot more air in order to temper the incoming air (required for comfort). This means that more fan power is required. Some specification of this fan power requirement should be used in Title 24 compliance calculations.

Appendix A: REGCAP model outline

Introduction

The REGCAP model combines a ventilation model, a heat transfer model and a simple moisture model.

The ventilation model developed here is a two zone model, in which the two zones are the attic and the house below it and they interact through the ceiling flow. Both zones use the same type of flow equations and solution method. The total building and attic leakage is separated into components and a flow equation is developed for each leakage site. The envelope flow components are illustrated in Figure 1.

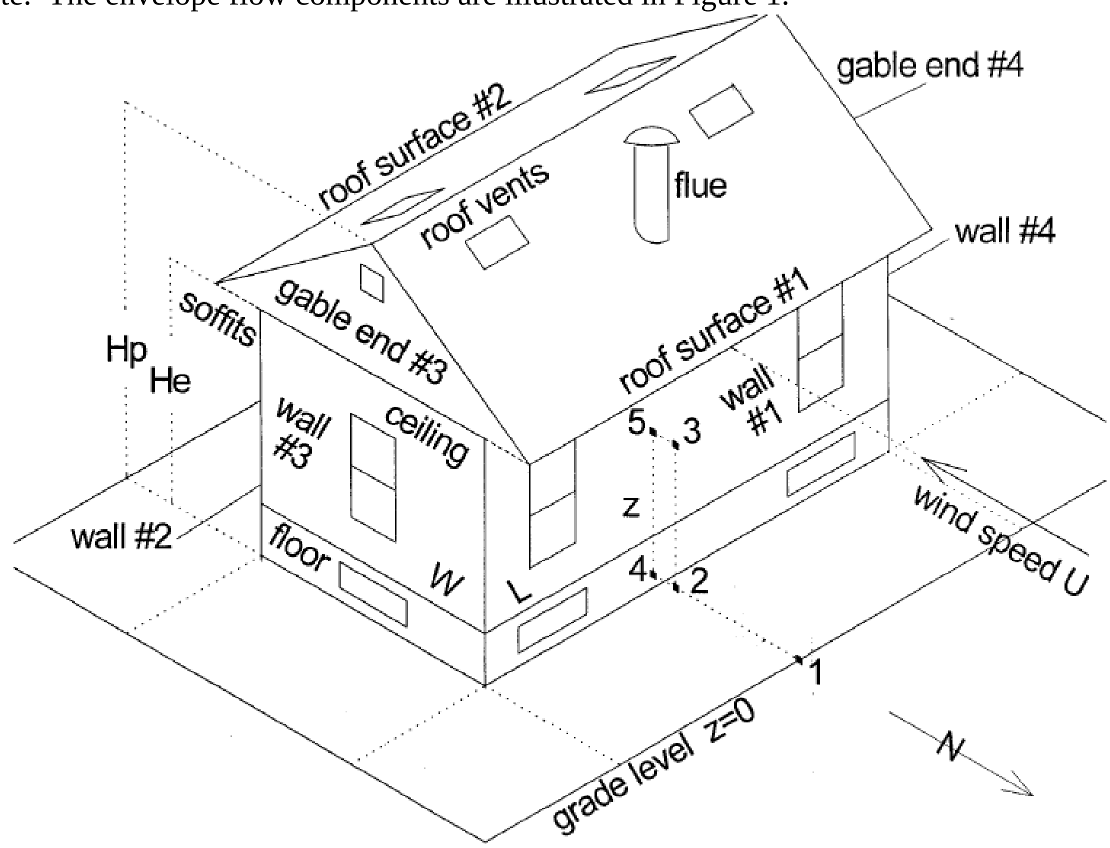

Figure 1. Illustration of house and attic air flow components

The flow at each leakage site is determined by a power-law pressure - flow relationship. This relationship has a flow coefficient, C, that determines the magnitude of the flow and an exponent for pressure difference, n, that determines how the flow through the leak varies with pressure difference. For each zone the total leakage is divided into distributed leakage that consists of the small cracks inherent in the building construction and intentional openings (e.g. furnace flues and open windows). Following the work of Sherman and Grimsrud (1980) the distributed envelope leakage is further divided into specific locations based on the height of the leak (i.e. floor, ceiling and walls). The building is assumed to have a rectangular planform with a user specified length, width and height. The attic has the same floor plan as the house and a pitched roof with soffits and gable ends.

In addition to the envelope leakage, the air flows in and out of attic ducts are included in the mass balances. The ducts are modelled differently depending on if the air handler is on or off. When the air handler is off, the duct leaks are assumed to experience the same pressure difference as the ceiling. Air then flows between the house and the attic via these leaks. When the air handler is on, supply leaks enter the attic and return leak flows are form the attic to the return duct and there are register flows between the ducts and the house.

The ventilation rate of the house and the attic is found by determining the internal pressures for the house and attic that balances the mass flows in and out. Because the relationship between mass flow and pressure is non-linear, the solution is found by iteration.

The attic heat transfer model determines the temperature of the attic air and the other components (e.g., pitched roof surfaces and ducts). A lumped heat capacity method is used to divide the attic into several nodes, and an energy balance is performed at each node to determine the temperatures. The attic air temperature is used to find the attic air density used in the ventilation calculations. The attic ventilation rate changes the energy balance for the attic air and the surface heat transfer coefficients. Fortunately this coupling of the attic ventilation model and the heat transfer model is weak because attic ventilation rates are not a strong function of attic air temperature.

A simple building load model is used to determine indoor air temperature. It uses the total UA for the building together with solar loads (including window orientation – i.e., the area of windows in facing north, south, east and west). A critical part of the house model is the coupling of the house air to the thermal mass of the structure and furnishings. The model uses a combination of thermal mass and surface area together with natural convection heat transfer coefficients.

An equipment model is used to determine heating and cooling system capacities, efficiencies and energy consumption. For gas or electric furnace heating the capacity is fixed for all conditions. For air conditioning, the indoor and outdoor air conditions, together with air handler flow and refrigerant charge are used to determined the cooling system performance.

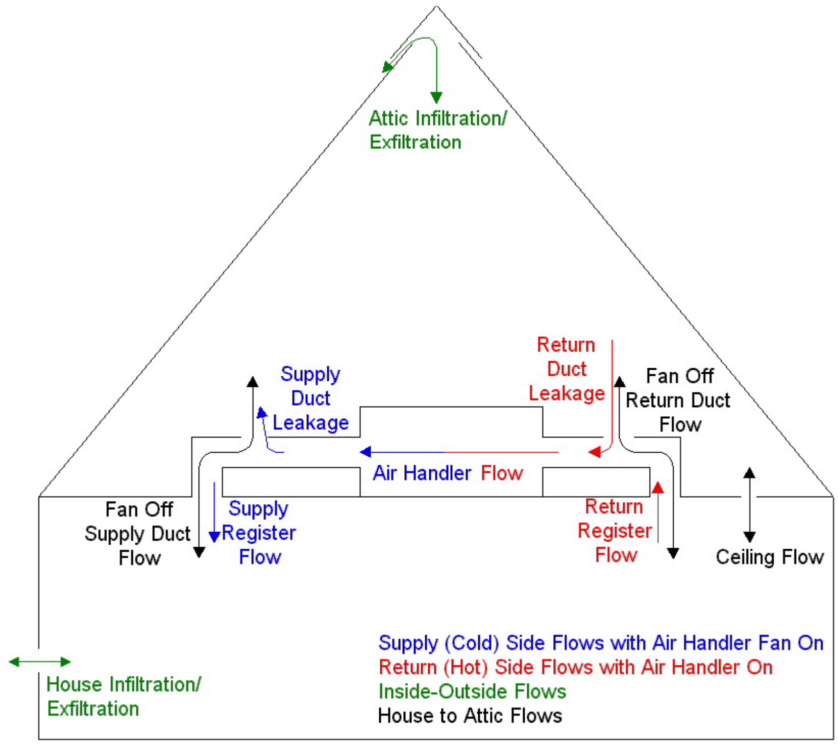

Figure 1: Schematic of duct related air flows (Arrows indicate direction). House and attic air infiltration/exfiltration is the sum of local and distributed leakage.

Ventilation model

The flow through each leakage path is found by determining the internal pressure in the house and attic that balances the mass flow rates. The house and attic interact through the pressure difference and flowrate through the ceiling and duct leaks, and the combined solution is found iteratively. The calculated ventilation rates are used as inputs to the heat transfer model and the building load model. The ventilation model and the heat transfer model are coupled because the ventilation rate effects the amount of outside and house air convected through the attic (as well as convective heat transfer coefficients) and the attic air temperature changes the attic air density. This change in density changes the mass flow rates and the stack effect driving pressures for attic ventilation. The combined ventilation and heat transfer model solution is found iteratively, with the ventilation rate being passed to the heat transfer model that then calculates an attic air temperature. This new attic air temperature is then used in the ventilation model to recalculate ventilation rates. The initial temperature estimate for the attic air used in the first iteration for the ventilation model is the outside air temperature. Most of the time the attic air is within a few degrees of the outside air temperature and the combined ventilation and heat transfer model requires only a few iterations (five or less).

Some significant limitations and assumptions for the ventilation model are listed below:

- There is assumed to be no valving action in the building and attic leakage so that flow coefficients are independent of flow direction.

- The building has a rectangular planform. The planform must not have the longest side greater than about three times the shorter side because the wind pressure coefficients used in the model will be incorrect.
- The attic has two pitched roof surfaces and gable ends. This assumption affects the leakage distribution and the pressure coefficients applied to the attic leakage sites.
- The interior of both the house and the attic are well-mixed zones.
- There are no indoor or outdoor vertical temperature gradients, so that the indoor and outdoor air densities are independent of location.
- Air behaves as an incompressible ideal gas. This allows density and viscosity to be functions of temperature only.
- Wall and pitched roof leakage is evenly distributed so as to allow simple integration of height dependent mass flow equations.
- All wind pressure coefficients are averaged over a surface. This means that extremes of wind pressure occurring at corner flow separations are not included.

General flow equation

The general flow equation for each leak is given by:

$$M = \rho C \Delta P^n \tag{1}$$

where M = Mass flow rate [kg/s]
ρ = Density of air flow [Kg/m^3]
C = Flow coefficient [$m^3/(sPa^n)$]
ΔP= Pressure difference across the leak [Pa]
n = Pressure exponent

The flow direction is determined by ΔP where a positive ΔP produces inflow and a negative ΔP produces outflow. A density and viscosity correction factor is applied to C to account for changes due to the temperature of the air flow.

Neglecting atmospheric pressure changes:

$$C = C_{ref}\left(\frac{T}{T_{ref}}\right)^{3n-2} \tag{2}$$

where T_{ref} is the absolute reference temperature (K) at which C_{ref} was measured, and T is the temperature of the airflow. For many buildings the distributed background leakage has n~2/3, which means that this correction is unity. For simplicity this temperature correction was therefore not applied to distributed leakage. For localised leakage sites including furnace flues, passive vents and attic vents n is typically 0.5 and this correction can become significant and therefore it is included in the ventilation calculations.

Each leak is then defined by its flow coefficient, pressure exponent, height above grade, wind shelter, and wind pressure coefficient. For distributed leakage on walls and pitched roof surfaces, an integral closed form equation is used. Similarly, for open doors and windows and integrated Bernoulli relation is used that includes interfacial mixing effects. For duct leakage with the air handler on, fixed user specified flow rate is used. For ventilation fans, a simple fan law is used so that the flow through the fan changes with the pressure difference across the fan. In the future these ventilation fan flows can simply be fixed values as the relationship between pressure difference and air flow is not generally known.

Wind Pressures

To find the outside surface wind pressure for each leak a wind pressure coefficient, Cp, is used that includes a windspeed multiplier, S_U to account for shelter. The wind speed, U, is the eaves height wind speed. The following equation is then used to calculate the pressure difference due to wind effect:

$$\Delta P_U = \rho_{out} Cp \frac{(S_U U)^2}{2} \tag{3}$$

where ΔP_U is the difference between the pressure on the surface of the building due to the wind and the atmospheric reference pressure P_∞ (at grade level, z=0). ρ_{out} is chosen as the reference density for pressures, because pressure coefficients are measured in terms of the external flow and the outdoor air density is used to calculate pressure coefficients from measured surface pressures. P_∞ is the pressure in the atmosphere far away from of the building where the building does not influence the flow field. S_U is a windspeed multiplier that accounts for windspeed reductions due to upwind obstacles. $S_U = 1$ implies no shelter and $S_U = 0$ implies complete shelter and there is no wind effect. Because each leak has a different Cp and S_U it is convenient to define a reference wind pressure P_U as

$$P_U = \rho_{out} \frac{U^2}{2} \tag{4}$$

and then Equation 3 can be written in terms of P_U:

$$\Delta P_U = Cp\, S_U^2 P_U \tag{5}$$

This definition is used later in the equations for the flow through each leak.

Indoor-Outdoor Temperature Difference Pressures

The hydrostatic pressure gradient inside and outside the building depends on the air temperature. Different temperatures inside and outside result in a differential pressure across the building envelope, ΔP_T. ΔP_T is defined as the outside pressure minus the inside pressure. This convention is applied so that positive pressures result in flow into the building (the same as for wind effect). Integrating the resulting pressure difference means that the stack effect pressure difference at height z above grade is given by

$$\Delta P_T(z) = -zg\,\rho_{out}\left(\frac{(T_{in} - T_{out})}{T_{in}}\right) \tag{6}$$

where z is the height above a reference (grade level) [m]
g is gravitational acceleration (9.81 [m/s^2]).

Each leak is at a different height, z , above grade, and so for convenience in writing the mass flow equations P_T is defined as follows:

$$P_T = g\,\rho_{out}\left(\frac{(T_{in} - T_{out})}{T_{in}}\right) \tag{7}$$

P_T is the pressure gradient and is multiplied by the height of each leak above grade to find the stack effect pressure difference at that location. Substituting Equation 6 in 5 gives:

$$\Delta P_T(z) = -z P_T \tag{8}$$

Total Pressure Difference

The total pressure difference is due to a combination of these wind and indoor-outdoor temperature difference effects, together with ventilation fan and HVAC system air flows, and the indoor to outdoor pressure shift (ΔP_I) that acts to balance the inflows and outflows. ΔP_I is the only unknown in this equation, and is the same for every leak in each zone. The total pressure difference is given by:

$$\Delta P = Cp S_U^2 P_U - z P_T + \Delta P_I \tag{9}$$

Equation 9 is applied to every leak for the building and the attic with the appropriate values of Cp, S_U and z.

The linear change in pressure, ΔP, with height, z, due to the stack effect term in Equation 9 means that when inflows and outflows are balanced there is a location where there is no pressure difference. This is called the neutral level, H_{NL}. For $T_{in} > T_{out}$ flow is in below H_{NL} and out above H_{NL}, and the flow directions are reversed for $T_{out} > T_{in}$. In general the neutral level is different for each wall due to the inclusion of wind pressures which can drive H_{NL} above the ceiling or below the floor. In those cases there is one way flow through the wall. The neutral level is found for the i^{th} vertical by setting $\Delta P = 0$ in Equation 9 and solving for $z = H_{NL,i}$:

$$H_{NL,i} = \left(\frac{\Delta P_I + S_{U,i}^2 Cp_i P_U}{P_T} \right) \tag{10}$$

Wind Pressure Coefficients For the house

Wind pressure coefficients are taken from wind tunnel tests and depend on the wind direction. For closely spaced houses in a row the pressure coefficients also change due to the change in flow around the building. Walker and Wilson (1994) discuss these vary in greater detail. Table 1 contains the wall averaged wind pressure coefficients used for the house by the ventilation model for wind perpendicular to the upwind wall. For the closely spaced row, the wind is blowing along the row of houses.

Table A1. Wall averaged wind pressure coefficients for a rectangular building with the wind normal to upwind wall from Akins, Peterka and Cermak (1979) and Wiren (1985).

Shelter Configuration	Cp, Wind Pressure Coefficient		
	Upwind Wall	Side Walls	Downwind Wall
Isolated House	+0.60	-0.65	-0.3
In-Line Closely-Spaced Row	+0.60	-0.2	-0.3

When the wind is not normal to the upwind wall an harmonic trigonometric function is used to interpolate between these normal values to fit the variation shown by Akins, Peterka, and Cermak and Wiren. For each wall of the building the harmonic function for Cp from Walker and Wilson (1994) is used:

$$Cp(\theta) = \frac{1}{2}[(Cp(1) + Cp(2))(\cos^2\theta)^{\frac{1}{4}} + (Cp(1) - Cp(2))(\cos\theta)^{\frac{3}{4}}$$
$$+(Cp(3) + Cp(4))(\sin^2\theta)^2 + (Cp(3) - Cp(4))\sin\theta] \qquad (11)$$

where Cp(1) is the Cp when the wind is at 0^o (+0.60)
Cp(2) is the Cp when the wind is at 180^o (-0.3)
Cp(3) is the Cp when the wind is at 90^o (-0.65 or -0.2)
Cp(4) is the Cp when the wind is at 270^o (-0.65 or -0.2)
and θ is the wind angle measured clockwise from the normal to the wall.
This function is shown in Figure 2 together with data from Akins et. al. for a cube. The error bars on the data points in Figure 2 represent the uncertainty in reading the measured values from the figures of Akins, Peterka and Cermak.

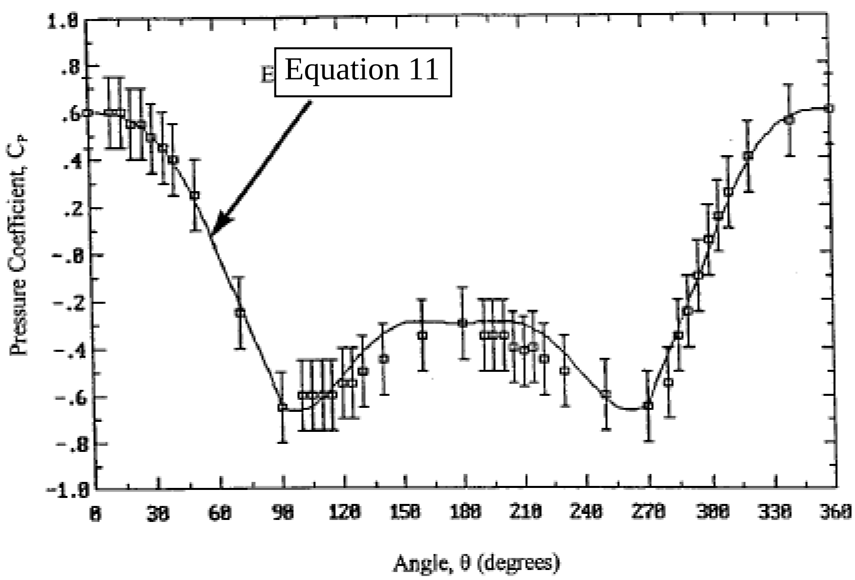

Figure 2. Angular variation in wind pressure coefficient for a rectangular building

Wind Pressure Coefficients For the Attic

The attic simulation model has been developed for a gable end attic with two pitched roof surfaces. The Cp's for gable ends or soffits are assumed to be the same as those on the walls below them and are calculated using the same procedure as for house walls. The pitched roof surfaces have Cp's that are also a function of roof slope. Table 2 gives values of Cp measured by Wiren (1985) for upwind and downwind pitched roof surfaces with wind normal to the upwind surface for different roof pitches. For wind flow parallel to the roof ridge Cp's change in the same way as for houses with Cp = -0.6 for an isolated building and Cp = -0.2 for row houses for both roof pitched surfaces. The Cp is independent of roof pitch for flow parallel to the roof ridge.

Table 2A. Pitched roof wind pressure coefficients for wind normal to the upwind surface (Wiren (1985))

Roof Pitch	Cp, Wind Pressure Coefficient	
	Upwind Surface	Downwind Surface
$<10^o$	-0.8	-0.4
10^o to 30^o	-0.4	-0.4
$>30^o$	+0.3	-0.5

To account for the variation on roof Cp with wind angle a similar empirical relationship to that for houses is used (from Walker, Forest and Wilson (1995)):

$$Cp(\theta) = \frac{1}{2}[(Cp(1) + Cp(2))\cos^2\theta + (Cp(1) - Cp(2))F$$
$$+(Cp(3) + Cp(4))\sin^2\theta + (Cp(3) - Cp(4))\sin\theta] \tag{12}$$

where Cp(1) is the Cp when the wind is at 0^o

Cp(2) is the Cp when the wind is at 180^o

Cp(3) is the Cp when the wind is at 90^o

Cp(4) is the Cp when the wind is at 270^o

θ is the wind angle measured clockwise from the normal to the roof surface.

F is a switching function to account for changes in roof pitch.

$$F = \frac{1-(|\cos\theta|)^5}{2}\left(\frac{28-\psi}{28}\right)^{0.01} + \frac{1+(|\cos\theta|)^5}{2} \tag{13}$$

where ψ is the roof pitch in degrees measured from horizontal. Equation 13 acts like a switch with F ~ 1 up to $\psi = 28^o$ and F ~ $\cos\theta$ when $\psi > 28^o$. The switch point of 28^o is chosen so that this relationship produces the same results as the wind tunnel data in Table 2. Equation 13 is not used to change the pressure coefficients shown in Table 2, but it changes the functional form of Equation 12 so that the interpolation fits the measured pressure coefficients.

Equation 12 is compared with pitched roof Cp's from Liddament (1986) in Figures 3 through 5 for roof pitches >30^o, 10^o to 30^o, and <10^o respectively.

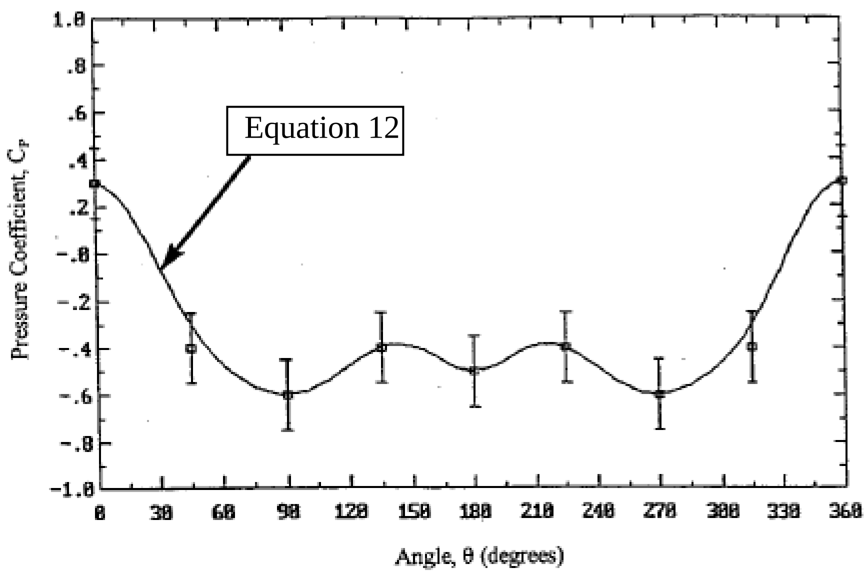

Figure 3. Roof pressure coefficients for a steep sloped roof (pitch > 30°)

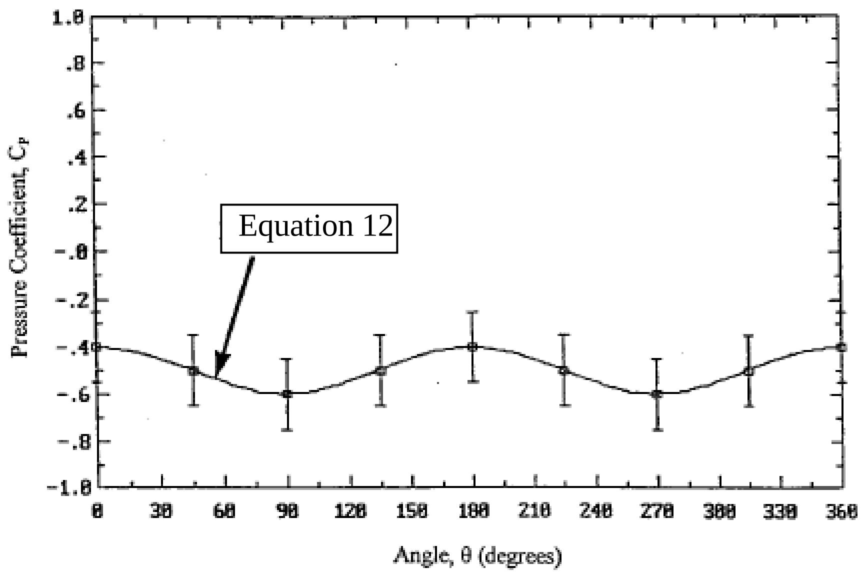

Figure 4. Roof pressure coefficients for a moderate sloped roof (10° < pitch < 30°)

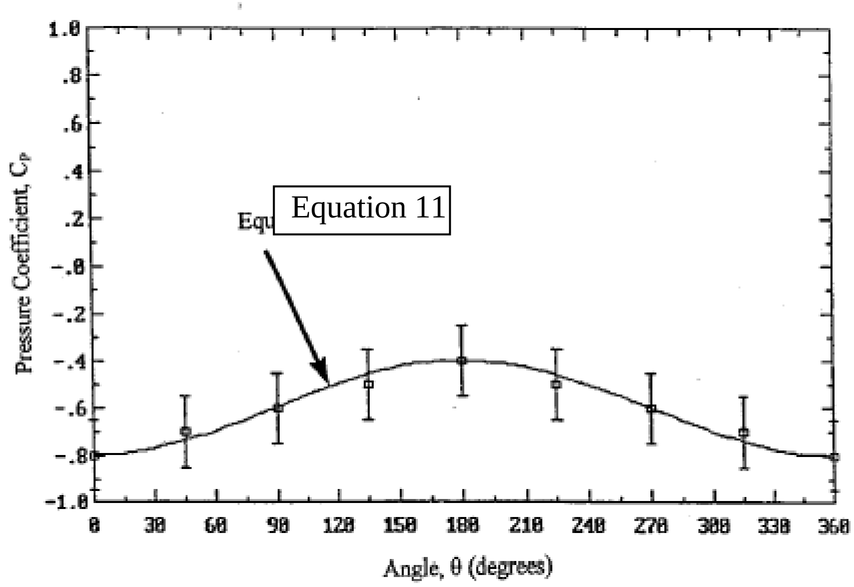

Figure 5. Roof pressure coefficients for a low sloped roof (pitch < 10°)

Wind Shelter

Shelter effects are separated from the effects of changing Cp's with wind direction and flow field changes. The windspeed multiplier, S_U, acts to reduce the effective windspeed generating surface pressures on the building such that:

$$U_s = S_U U \tag{14}$$

where U is the free stream windspeed with no sheltering effects.

S_U has the limits where $S_U = 1$ implies no shelter and $S_U = 0$ implies total shelter and there are no wind pressures on the building.

U_S is the effective windspeed used for calculating surface pressures.

The coefficients used to find U_S and S_U are based on measured surface pressures and not on measured wake velocities.

REGCAP has the following three options for wind shelter.

1. Fixed shelter for all wind directions.

2. Interpolation Function.

The interpolation function determines shelter for all wind angles given shelter for four cardinal directions so that for each wall:

$$S_U = \frac{1}{2}[(S_U(1) + S_U(2))\cos^2\theta + (S_U(1) - S_U(2))\cos\theta$$
$$(S_U(3) + S_U(4)\sin^2\theta + (SU(3) - S_U(4))\sin\theta] \qquad (15)$$

where S_U is the windspeed multiplier

$S_U(1)$ is the S_U when the wind is at 0^o

$S_U(2)$ is the S_U when the wind is at 180^o

$S_U(3)$ is the S_U when the wind is at 90^o

$S_U(4)$ is the S_U when the wind is at 270^o

and θ is the wind angle measured clockwise from the normal to the upwind wall.

3. Input from data file:

A file of shelter values for every degree of wind direction for all four faces of a house was generated using sophisticated wind shelter calculation techniques discussed in Walker, Wilson and Forest (1996). The following figure illustrates the values of shelter coefficient in the pre-calculated data file for one wall.

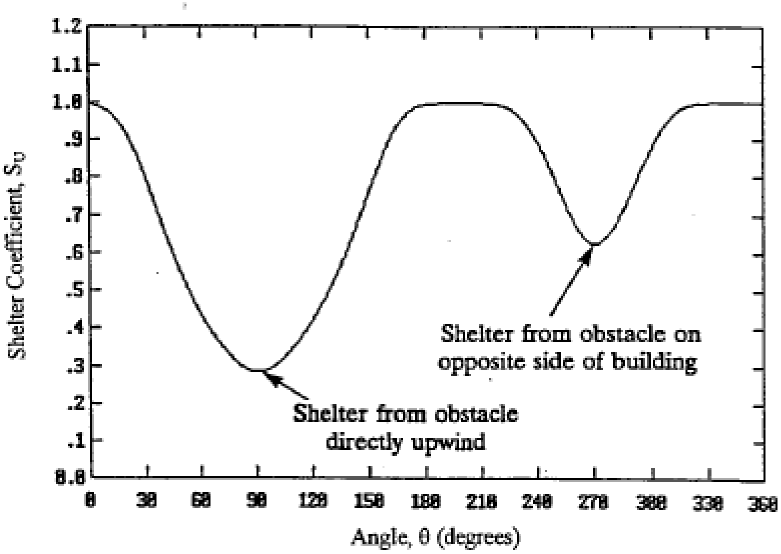

Figure 6. Wind shelter for a typical urban house

Flow Through each Leak for the Attic

The total leakage is divided into distributed leakage and localised leakage. All the distributed leakage sites are assumed to have the same flow exponent. The flow coefficients for the roof and soffit must be estimated as fractions of the total distributed leakage such that

$$C_{d,a} = \sum_{i=1}^{4} C_{s,i} + C_r \tag{16}$$

where C_r is the total leakage in the two pitched roof surfaces and $C_{s,i}$ is the leakage in the soffit or gable ends above each wall.

Pitched roof Leakage

The two pitched roof surfaces are assumed to have equal leakage. Therefore there is $C_r/2$ leakage in each surface. Cp for the pitched roof surfaces is found using Equation 12 and Table 2. If the surrounding obstacles are taller than the building in question then S_U for the pitched roof surfaces is estimated to be the same as the wall below them, otherwise there is no shelter and $S_U=1$. For example, a south facing roof pitch would then have the same S_U as calculated for the south facing wall below it. For the attic roof the neutral level, $H_{NL,r}$, is calculated for the two roof pitches using the appropriate Cp and S_U values in Equation 10.

The change in pressure with height, z, on the roof surfaces makes the flow through the roof a function of height which must be integrated to find the total mass flow in and out of each roof surface $M_{r,i}$.

$$M_{r,i} = \int dM_{r,i}(z)dz \tag{17}$$

where

$$dM_{r,i}(z) = \rho\, dC_{r,i}(\Delta P_{r,i}(z))^{n_r} \tag{18}$$

where $\Delta P_{r,i}(z)$ is given by Equation 9. Assuming evenly distributed leakage allows easy integration over the roof because the fractional leakage $dC_{r,i}$ is given by:

$$dC_{r,i} = C_{r,i}\frac{dz}{(H_p - H_e)} \tag{19}$$

where H_p is the roof peak height and H_e the eave height. Substituting Equations 19 and 18 in 17 gives

$$M_{r,i} = \frac{\rho C_{r,i}}{(H_p - H_e}\int \Delta P_{r,i}^{n_r} dz \tag{20}$$

where the limits of integration depend on the neutral level height, $H_{NL,r}$, that is found for each wall using Equation 18.

When $H_{NL,r}$ is on the roof there is flow both in and out of the roof and upon integrating Equation 20 the masses flowing in and out are kept separate. This important for the total mass balance and for keeping track of all the flows through the building envelope. There are several different cases of flow through the pitched roof surfaces depending on the location of $H_{NL,r}$, T_a and T_{out}. The pressure differences at the eave height, ΔP_e, and at the roof peak, ΔP_p, are defined as follows and are convenient to use when calculating the mass flow rates.

$$\Delta P_p = \Delta P_{I,a} + S_U^2 Cp\, P_U - H_p\, P_{T,a} \tag{21}$$

$$\Delta P_e = \Delta P_{I,a} + S_U^2 Cp\, P_U - H_e\, P_{T,a} \tag{22}$$

An example case given by Equations 23 and 24 is for $T_a > T_{out}$ with $H_{NL,r}$ somewhere on the pitched roof surface between the eave height, H_e, and the peak height H_p. There is two way flow through the roof surface in this case with flow in below $H_{NL,r}$ and flow out above $H_{NL,r}$:

$$M_{r,out} = \frac{\rho_a \frac{C_r}{2} \Delta P_p^{(n_r+1)}}{(H_p - H_e)\, P_{T,a}\, (n_r + 1)} \tag{23}$$

$$M_{r,in} = \frac{\rho_{out} \frac{C_r}{2} \Delta P_e^{(n_r+1)}}{(H_p - H_e)\, P_{T,a}\, (n_r + 1)} \tag{24}$$

Soffit and Gable Leakage

The soffit and gable leakage are treated identically. The soffit and gable leakage is split into four parts, one for each side of the building. $C_{s,i}$ is the estimated fraction of the total attic distributed leakage in the soffit or gable on the i^{th} side of the building. H_s is the height of the leakage above grade and usually $H_s = H_e$ for soffits. For the gable leakage H_s is assumed to be H_e plus half of the attic height (H_p - H_e). The wind pressure coefficient (Cp_i) and shelter factor ($S_{U,i}$) are assumed to be the same as for the wall below each soffit or gable. The pressure difference across each soffit or gable above wall i is then given by:

$$\Delta P_{s,i} = \Delta P_{I,a} + Cp_i\ S_{U,i}^2\ P_U - H_s P_{T,a} \tag{25}$$

Attic Vent Leakage

Attic vents provide extra ventilation leakage area in addition to the background distributed leakage. There can be multiple attic vents at different locations on the attic envelope, each with their own C_V and n_V. C_V and n_V are user specified leakage characteristics of each vent. Usually the vent can be assumed to act like an orifice with $n_V = 0.5$. In that case C_V can be estimated from the vent area multiplied by the discharge coefficient, K_D. The vent area should be corrected for any blockage effects e.g. by insect screens. $S_{U,V}$ and Cp_V for each vent are the same as for the attic surface they are on, either the gable ends (which have the same S_U and Cp as the wall below them) or the roof pitches. H_V is the height above grade of the vent and the pressure difference across each attic vent is given by:

$$\Delta P_{V,a} = \Delta P_{I,a} + S_{U,V}^2\ Cp_V\ \ P_U - H_V\ \ P_{T,a} \tag{26}$$

$\Delta P_{V,a}$ is calculated for each attic vent and the flow through each attic vent is given by Equation 1.

Attic Floor Leakage

The mass flow rate through the attic floor is calculated by the house zone part of the ventilation model. The resulting $\Delta P_{I,a}$ from balancing the mass flows for the attic zone is returned to the house zone to calculate pressure across the ceiling, and then to recalculate the mass flow through the attic floor.

Ventilation Fans in Attics

Fans are included by using a fan performance curve. The operating point on the curve is determined by the pressure across the fan. The stack and wind pressures across each fan are found by specifying which attic surface the fan is located in and its height above grade, H_{fan}. Cp_{fan} and $S_{U,fan}$ are the same as the surface the fan is are located in. There can be multiple fans each with their own rated flowrates, Q_{rated}, and rated pressure differences, ΔP_{rated}. The pressure difference across each attic fan, $\Delta P_{fan,a}$, is given by:

$$\Delta P_{fan,a} = \Delta P_{I,a} + S_{U,fan}^2 Cp_{fan} P_U - H_{fan} P_{T,a} \tag{27}$$

Approximating the fan performance curve by a power law using p_{fan} gives the following equation for mass flow through each fan:

$$M_{fan,a} = \rho Q_{rated} \left(\frac{\Delta P_{rated} + \Delta P_{fan,a}}{\Delta P_{rated}} \right)^{p_{fan}} \tag{28}$$

where ρ is equal to ρ_a for outflow and ρ_{out} for inflow.

Duct Leaks with air handler off (M_{soff} and M_{roff})

Both the supply and return leaks have the same pressure difference as the attic floor/house ceiling. The supply leakage pressure exponent is a required input, but typically a value of 0.6 is used. The flow coefficient is calculated from the leakage air flow rate, assuming a refernce pressure of 25 Pa and using the pressure exponent:

$$C_{soff} = \frac{Q_{ah}\alpha_s}{25^{n_s}} \tag{29}$$

$$C_{roff} = \frac{Q_{ah}\alpha_r}{25^{n_r}} \tag{30}$$

where, C_{soff} is the supply leak flow coefficient, Q_{ah} is the air handler flow, n_s is the supply leak pressure exponent and α_s is the supply leakage expressed as a fraction of air handler flow. C_{roff} is the return leak flow coefficient, n_r is the return leak pressure exponent and α_r is the return leakage expressed as a fraction of air handler flow.

Duct leaks with air handler on (M_{son} and M_{ron})

All the air handler on flows: air handler flow, duct leakage flows and register flows are converted from the input volumetric flows to mass flows using the indoor air density. The supply leak mass flow is added to the inflow into the attic and the return leaks are treated as air flows out of the attic. These are fixed mass flows independent of wind, stack or internal pressures and simply appear as mass flows in the mass balance equation.

Flow through Each Leak for the House

The flow coefficients for the ceiling, floor level leaks and walls are estimated as fractions of the total distributed leakage such that

$$C_d = \sum_{i=1}^{4} C_{f,i} + \sum_{i=1}^{4} C_{w,i} + C_c \tag{31}$$

where $C_{f,i}$ is the floor level leakage below wall i, $C_{w,i}$ is the leakage in wall i and C_c is the ceiling leakage.

Furnace Flues and Fireplaces

Furnace flues and fireplaces are usually the largest openings in the building envelope and typically have a flow exponent, n_F, close to 0.5. The flue leakage coefficient, C_F, can be calculated from diameter, D_F, of the flue or fireplace assuming orifice flow, with a discharge coefficient of $K_D = 0.6$. The pressure coefficient of $Cp_F = -0.5$ is from Haysom and Swinton (1987). The change in wind velocity with height above grade may be significant for furnace flues that protrude above the reference eaves height. A corrected Cp_F is then given by:

$$Cp_F = (-0.5)\left(\frac{H_F}{H_e}\right)^{2p} \tag{32}$$

where H_f is the flue top height and p is the exponent used in the atmospheric boundary layer wind profile (typically p=0.3 for urban surroundings and p=0.17 for rural sites). Shelter for the flue, $S_{U,F}$, is the shelter factor at the top of the flue. If the surrounding buildings and other obstacles are below the flue height then it is assumed that $S_{U,F} = 1$. If the surrounding obstacles are higher than the flue then the flue is sheltered and $S_{U,F}$ is calculated using Equation 15. The general pressure difference Equation 8 can be written specifically for the furnace flue as:

$$\Delta P_F = \Delta P_I - P_T \ H_F + P_U \ S_{U,F}^2 \ Cp_F \tag{33}$$

and the mass flow rate, M_F, for the flue is given by Equation 1. Note that this is for an unheated flue or a natural draft furnace (flue without a draft inducing fan) well connected to the conditioned space. In new construction most furnace flues will be outside conditioned space in a well vented closet, garage or attic (or will be direct vented), in which case the flue leak is set to zero and only open fireplaces need to be considered, or we need to know the flow rate through the forced combustion fan for furnaces.

Floor Level Leakage

The leakage at floor level, $C_{f,i}$, is estimated as a fraction of the total distributed leakage and n_f is the same as n for the other distributed leaks. There are two cases of floor level leakage that require different assumptions about wind pressure effects. The cases depend on house construction.

a. Basements and Slab on Grade

In this case the total floor level leakage is split into four parts, one for each side of the building. On each side the floor level leakage is given the same Cp and S_U as the wall above it. For the i^{th} side of the building

$$\Delta P_{f,i} = \Delta P_I + Cp_i \; S_{U,i}^2 \; P_U - H_f \; P_T \tag{34}$$

where floor height, H_f, is measured from grade level. For a house with a basement this is the height of the main level floor above grade and the leakage coefficient, $C_{f,i}$ includes the leakage around basement windows, dryer vents etc. The mass flow rate for these floor level leaks is given by Equation 1.

b. Crawlspaces (flow through house floor to and from the crawslpace)

As an estimate of the wind pressure in a crawl space the shelter and pressure coefficients for the four walls of the building are averaged. The average is weighted for non square plan buildings by the length of each side, L_i, so that for the i^{th} side.

$$Cp_f = \sum_{i=1}^{4} S_{U,i}^2 Cp_i \left(\frac{L_i}{L_\pi} \right) \tag{35}$$

where L_π is the perimeter of the building (the sum of the L_i's) and then the pressure across the crawlspace is given by

$$\Delta P_f = \Delta P_I + Cp_f P_U - H_f P_T \tag{36}$$

and the mass flow rate through the crawlspace leakage is given by Equation 1.

Ceiling Leakage

The ceiling flow coefficient C_c is estimated from the total distributed leakage and n_c is the same as n for the other distributed leaks. There are no wind pressures acting on the ceiling except indirectly through the flow balancing pressures ΔP_I (house) and $\Delta P_{I,a}$ (attic) because the ceiling is completely sheltered from the wind. The pressure across the ceiling includes the difference in attic and house buoyancy pressures

$$\Delta P_c = \Delta P_I - \Delta P_{I,a} - \rho_{out} g H_e \left(\frac{T_{in} - T_{out}}{T_{in}} - \frac{T_a - T_{out}}{T_a} \right) \tag{37}$$

The mass flow rate through the ceiling is given by Equation 1.

Wall Leakage

For each wall $C_{w,i}$ is estimated from the total distributed leakage and the flow exponent, n, for each wall is n_d, the same as for the other distributed leaks. The vertical distributed leakage is treated the same way as attic pitched roof leakage.

Fan Flow

Fans are included in houses the same was as for attics: by using the naturally occurring pressures to determine the operating point on a fan curve.

Vent Leakage

The vent leakage is attributed to deliberately installed leakage sites that are separate from the background leakage. Multiple vents can be described, each with their own flow characteristics and each at a different location on the house envelope. Furnace and fireplace flues are treated separately as they may contain heated air that would produce a different stack effect for that leak only. Vents exiting through the roof use the same Cp and S_U as the furnace flue. The pressure difference and mass flow calculation is the same as for attic vents.

Flow through open Doors and Windows

The flowrates through door and window openings are determined by integrating the flow velocity profiles found by applying Bernoulli's equation along streamlines passing through the opening as shown by Kiel and Wilson (1986). For convenience the following parameters are defined

$$P_b = Cp\, S_U^2 U^2 - 2g\, H_b \left(\frac{T_{in} - T_{out}}{T_{in}} \right) + \frac{2\Delta P_I}{\rho_{out}} \tag{38}$$

$$P_t = Cp\, S_U^2 U^2 - 2g\, H_t \left(\frac{T_{in} - T_{out}}{T_{in}} \right) + \frac{2\Delta P_I}{\rho_{out}} \tag{39}$$

where Cp and S_U are for the surface that the opening is in

H_b = Height above grade of the bottom of the opening

H_t = Height above grade of the top of the opening

As with the integrated wall flows the mass flows in and out depend on H_{NL}, T_{in} and T_{out}. All of the possible cases for flow above and below H_{NL} are given in appendix A. Appendix A also contains a derivation for the flow in below H_{NL} for the case where H_{NL} falls in the opening and $T_{in}>T_{out}$, such that

$$M_{out} = (\rho_{out}\rho_{in})^{\frac{1}{2}} \frac{KWT_{in}}{3g(T_{in} - T_{out})} P_t^{\frac{3}{2}} \tag{40}$$

$$M_{in} = \rho_{out} \frac{KWT_{in}}{3g(T_{in} - T_{out})} P_b^{\frac{3}{2}} \tag{41}$$

Window and Door Flow Coefficient, K

The flow coefficient, K, accounts for reduction in flow due to flow contraction, viscous losses and interfacial mixing. An estimate for K that accounts for the variation in

K due to interfacial mixing generated by atmospheric turbulence is given by Kiel and Wilson (1986) as

$$K = 0.400 + 0.0045 \, | T_{in} - T_{out} | \tag{42}$$

The flow coefficient must be altered when the interface is near the top or the bottom of the opening so that the iterative solution of flow for the whole building does not have the neutral level oscillating just above and below the top or bottom of the opening. A first order approximation is to let K vary linearly in the top and bottom 10% of the opening between the value of K with the neutral level at 10% or 90% of the opening height and K = 0.6 at the edges of the opening. This is physically realistic because when the interface is near the top or the bottom of the opening the edges of the opening will interfere with the interfacial mixing process. This will make the flow look more like one way flow with an assumed orifice discharge coefficient, $K_D = 0.6$.

Grille Air Flows

The supply and return grille air flows are determined by subtracting the leakage from the air handler flow. The volumetric flows are converted to mass flows using the indoor air density.

Air Flow Solution Method

All of the flow equations for the house contain the difference between the inside and outside pressure, ΔP_I, that is the single unknown (or $\Delta P_{I,a}$ for the attic). To find ΔP_I all of the flow equations are combined into one equation that is the mass balance for air in the house:

$$\sum M = M_F + M_f + M_c + M_{\sup} + M_{ret} + M_{son} + M_{soff} + \sum_{i=1}^{4} M_{ws,i} + M_v + M_{fan} = 0 \quad (43)$$

where the various mass flows are:
M_F : Flue
M_f : floor level leaks
M_c : Ceiling
M_{sup} : supply register air handler on
M_{ret} : return register air handler on. For CFI systems the CFI supply flow is subtracted from M_{ret}.
M_{soff} : supply register air handler off
M_{roff}: return register air handler off
M_V : sum of all passive vent flows
M_{fan} : is the sum of all the ventilation fans.

This equation for mass balance is non-linear with ΔP_I as the only unknown. To solve for ΔP_I, an iterative bisection technique was adopted because it is extremely robust and computational simple. This bisection search technique assumes that $\Delta P_I = 0$ for the first iteration and the mass inflow or outflow rates are calculated for each leak. At the next iteration ΔP_I is chosen to be +25 Pa if total inflow exceeds total outflow and -25 Pa if outflow exceeds inflow. Succeeding iterations use the method of bisection in which ΔP_I for the next iteration is reduced by half the difference between the last two iterations, thus the third iteration changes ΔP_I by ±6.25 Pa. The sign of the pressure change is positive if inflow exceeds outflow and negative if outflow is greater then inflow. The limit of solution is determined by stopping when the change in ΔP_I is < 0.01 Pa, which gives mass flow imbalances on the order of 0.001 Kg/s (or 4Kg/hour) for a typical house.

For the attic the mass balance equation is given by

$$\sum M = M_r + M_c + M_{soff} + M_{roff} + M_{son} + M_{soff} + \sum_{i=1}^{4} M_{s,i} + M_{v,a} + M_{fan,a} = 0 \qquad 1\mathbf{(44)}$$

where
M_r : sum of the in and the out flows through the pitched roof surfaces,
Mc : ceiling
M_{soff} : supply leak air handler off
M_{roff}: return leak air handler off
M_{son} : supply leak air handler on
M_{ron} : return leak air handler on
$M_{s,i}$ flow through soffit (or gable) component i.
$M_{fan,a}$: sum of the mass flows through all the attic fans

$M_{V,a}$: sum of the flows through all the attic vents.

As with the house all of the components of this mass balance equation contain the single unknown, $\Delta P_{I,a}$, the attic to outdoor pressure difference. The attic zone is solved using the same bisection technique as the house zone.

Zone Coupling

The house and attic zones are coupled by the flow through the ceiling and pressure difference across the ceiling. The house zone uses $\Delta P_{I,a}$ to calculate the mass flow through the ceiling. This mass flow is used in the mass flow balance by the attic zone to calculate a new $\Delta P_{I,a}$. This is an iterative procedure that continues until the change in mass flow through the ceiling from iteration to iteration is less than the magnitude of the house leakage coefficient divided by 10 or 0.0001 kg/s, whichever is larger.

Heat Transfer Model

A standard lumped heat capacity analysis is used, and solid material use the standard technique of splitting into surface and inner layers. The surface layer interacts by convection and radiation and the inner layer by conduction to the surface. The north and south sheathing are separated so that they may have different daytime solar gains. Forced convection heat transfer coefficients are used inside the attic using air flows calculated in the ventilation model. Radiation heat transfer inside the attic is simplified to three attic surface nodes: the attic floor and the two pitched roof surfaces plus the supply and return duct surfaces.

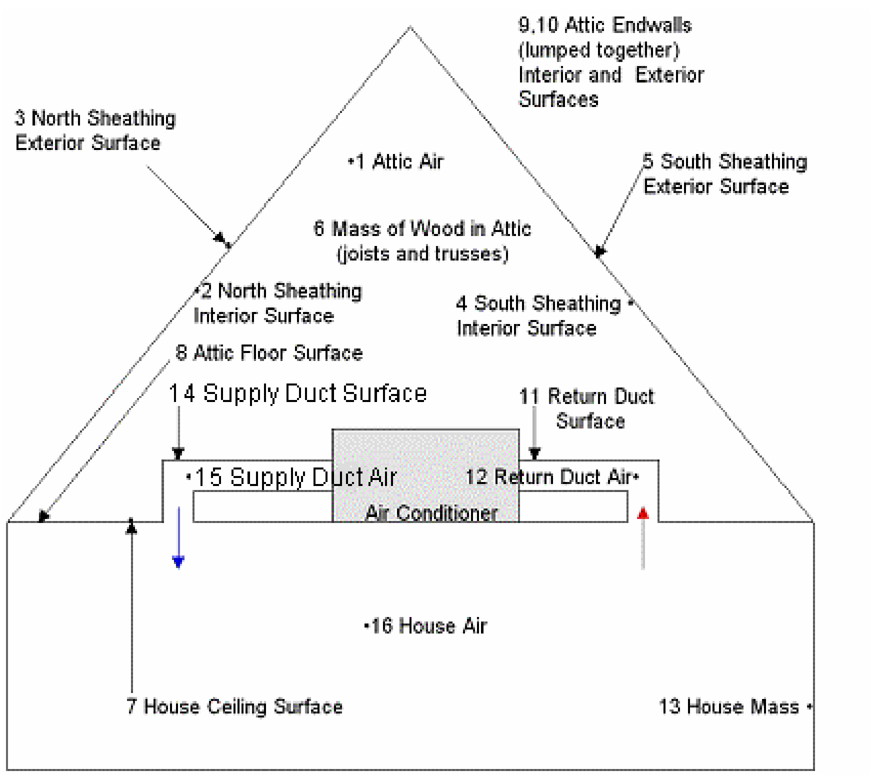

Figure 7: Nodes For Heat Transfer Model

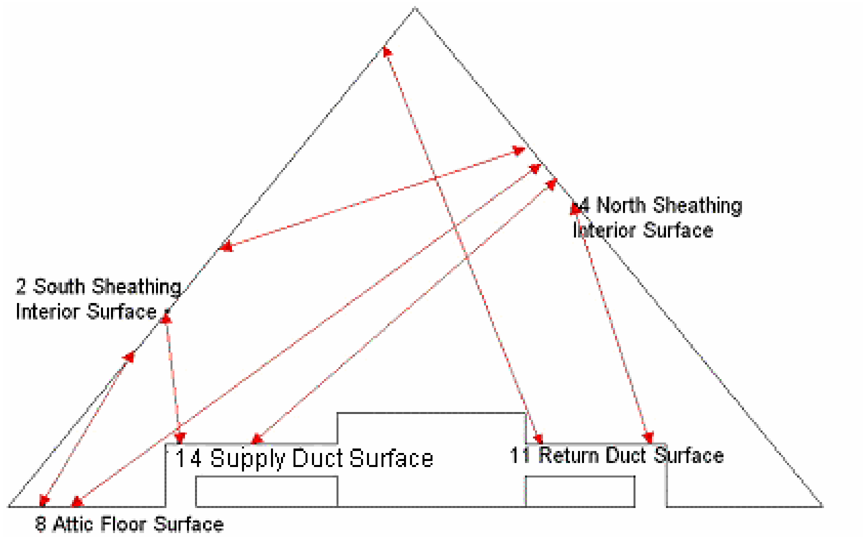

Figure 8: Radiation Transfer for Ducts in Attic

Attic and duct system heat transfer nodes:
1 = Attic Air
2 = Inner Surface1 Sheathing
3 = Outer Surface1 Sheathing
4 = Inner Surface2 Sheathing
5 = Outer Surface2 Sheathing
6 = All of the wood (joists, trusses, etc) lumped together
7 = Ceiling of the house
8 = Floor of the attic
9 = Inner End Wall
10 = Outer End Wall
11 = Return Duct Outer Surface
12 = Return Duct Air
13 = Mass of the house
14 = Supply Duct Outer Surface
15 = Supply Duct Air
16 = House Air

At each node, the rate of change of energy is equal to the sum of the heat fluxes

$$\rho_i V_i C_{sh,i} \frac{dT_i}{dt} = \sum q \qquad (45)$$

where ρ_i is the density [Kg/m^3] , V_i is the volume [m^3], $C_{sh,i}$ is the specific heat [J/KgK], T_i is temperature [K] and q are the heat fluxes [W]. The fluxes are due to convection, radiation and conduction heat transfer. The derivative in this equation is calculated using a finite difference approximation. Only the first term of the finite difference approximation is used so that the equation remains linear with temperature.

$$\rho_i V_i C_{sh,i} \frac{T_i^{j} - T_i^{j-1}}{\tau} = \sum q \tag{46}$$

were j refers to the current timestep and j-1 the previous timestep and τ is the length of the time step. The energy balance is performed at each timestep j with the previous hour's (j-1) temperatures used to calculate the rate of change of energy at each node. This results in a linear system of 16 equations and 16 unknowns (the temperatures) that can be solved using simple matrix solutions.

Radiation Heat Transfer
Inside the Attic (Nodes 2,4, 8, 11 and 14)

For simplicity, this model assumes that the radiation heat transfer inside the attic can be simplified to five surfaces: attic floor, two pitched roof sections plus the supply and return duct surfaces. The calculation of radiation exchange inside the attic is based on heat exchange between non-blackbodies.

$$q_i = A_i h_{R,i-j}\left(T_i - T_j\right) + A_i h_{R,i-k}\left(T_i - T_k\right) \tag{47}$$

where $h_{R,i\text{-}j}$ are radiation heat transfer coefficients from node i to node j that are calculated from

$$h_{R,i-j} = \frac{\sigma\left(T_i + T_j\right)\left(T_i^2 + T_j^2\right)}{\frac{1-\varepsilon_i}{\varepsilon_i} + \frac{1}{F_{i-j}} + \frac{\left(1-\varepsilon_j\right)A_i}{\varepsilon_j A_j}} \tag{48}$$

where ε = emissivity of surface, A is the area of the body and σ is the Stephan-Boltzman constant that is equal to $5.669*10^{-8}$, and $F_{i\text{-}j}$ are the view factors (see Appendix B). These equations represent a linearized solution to the radiant heat transfer between three bodies: i, j and k.

The emissivity of surfaces found in building construction is given by ASHRAE (1989)(Chapter 37). For the inside sheathing surfaces a typical value for wood is $\varepsilon = 0.90$ and for the attic floor that is assumed to be covered with fibreglass insulation the typical emissivity glass (from ASHRAE (1989), Chapter 37) is used, $\varepsilon = 0.94$. The emissivity of glass is also typical of diffuse surfaces, and the fibreglass insulation is a diffuse surface due to its roughness. The geometry factors are determined from the attic dimensions and duct locations. For example, for ducts on the attic floor, it is assumed that 1/3 of the duct surface area sees each pitched roof surface and the remaining third of the duct surface area is not involved in radiation heat transfer.

Solar Radiation (Nodes 3 and 5)

Solar gains are only applied to the external sheathing surfaces. The energy transfer due to solar radiation is

$$q_R = A\alpha G \tag{49}$$

where q_R is radiation heat transfer rate [W]
A = Surface area [m^2]

α = Surface absorbtivity

G = Total Solar Radiation [W/m^2], both direct and diffuse.

The radiant heat transfer properties (and thermal resistance) change depending on the attic sheathing material either: asphalt shingles (R=0.077, ε=0.91 α=0.92), white coated asphalt shingles(R=0.077, e=0.91 α=0.15), red clay tile (R=0.5, ε=0.58 α=0.92) and low emissivity coated clay tile(R=0.5, ε=0.5 α=0.92).

Radiant Exchange of Exterior Surfaces with Sky and Ground (Nodes 3 and 5)

In addition to the daytime solar gain the outside of the pitched roof sheathing has low temperature long wave radiant exchange with the sky and the ground. This exchange is responsible for cooling of the sheathing at night as it radiates energy to the cooler sky. On a cloudy night the cooling of the sheathing is reduced because the radiation exchange is with clouds that are warmer than the sky temperature. Both the clouds and the ground are assumed to be at the outside air temperature. The view factors that account for the proportion of sky, cloud or ground seen by the pitched roof surface are from Ford (1982). Cloud cover is taken from the WYEC2 CEC ACM weather data files (Total Sky Cover).

The net radiation exchange for exterior pitched roof sheathing surfaces has the same form as Equations 41 and 42 for the internal radiation because this is a three body problem involving the roof surface, the sky and the ground and the clouds (which are assumed to be at the same temperature). The sky temperature T_{sky} depends on the water vapour pressure in the air. The view factors give the fraction of exposure to the ground (and clouds) and the sky for the pitched roof surfaces. Using the same view factors for both pitched roof surfaces assumes that the cloud cover is uniformly distributed over the sky.

Effective Sky Temperature for Radiation

The sky temperature, T_{sky}, is the equivalent temperature of an imaginary blackbody that radiates energy at the same rate as the sky. The effective sky temperature, T_{sky}, is a function of air temperature, T_{out}, and water vapour pressure P_v. Parmelee and Aubele (1952) developed the following empirical fit to measured data to estimate T_{sky} for horizontal surfaces exposed to a clear sky.

$$T_{sky} = T_{out}\left(0.55 + 5.68\times10^{-3}\sqrt{P_v}\right)^{0.25} \tag{50}$$

where P_v is in Pascals and the temperatures are in Kelvin. Sample calculations show how T_{sky} can be very different from T_{out}. For example at T_{out} = 273K and 50%RH (so that P_v = 305 Pa) then T_{sky} = 245K, almost 30K difference.

Radiant Exchange of the Ceiling (Node 7) with the Room Below

This is modelled as a two body enclosed system where one body is the ceiling and the other body is the interior surfaces. The interior surfaces are assumed to be all at the same temperature as the inside air, T_{in}. The same linearization as for the pitched roof surfaces and the attic floor is applied so that the radiation heat transfer, $q_{R,7}$, is a linear function of temperature. The heat transfer coefficient is calculated based on the previous timestep temperatures.

Convection Heat Transfer

Natural and forced convection heat transfer coeeficients are calculated based on surface temperatures and local air velocities. The natural convection heat transfer is given by

$$q_T = h_T A \Delta T \tag{51}$$

where q_T is the free convection heat transfer rate [W]

h_T is the free convection heat transfer coefficient [W/m^{2}K] – this is given a fixed value of 3.2 based on a heated plate facing upwards.

A is the surface area

ΔT is the temperature difference

$$h_T = 3.2\left(\Delta T\right)^{\frac{1}{3}} \tag{52}$$

To keep the heat transfer equations linear, ΔT is evaluated using the previous hours temperatures.

The house ceiling uses a convection heat transfer coefficient of 6 W/m^{2}Kwith the air handler off (based on values in ASHRAE Fundamentals 200, Chapter 3) and 9 W/m^{2}K with the air handler on (based on typical indoor air velocities). These same heat transfer coefficients are used for the exterior surfaces of ducts when they are inside the conditioned space.

Forced Convection

Forced convection heat transfer is calculated using:

$$h_{forced} = (18.192 - 0.037T_{film})U^{0.8} \tag{53}$$

The constants are based on Nusselt correlations and the velocity, U, is based on local air velocities. For duct interior surfaces this is the average duct air velocity. For outside nodes (pitched roof surfaces) it is based on the windspeed. For the inside of attics a characteristic velocity is calculated based on attic envelope air leakage rates and the attic leakage area:

$$U_{atticconvection} = \frac{\left(Matticenvin - Matticenvout\right)}{\rho_{attic}\, 4Al_4} \tag{54}$$

Where Al_4 is the four pascal attic leakage area. Note that Matticenvout will be a negative number hence the subtraction sign.

For the interior and exterior attic surfaces, the natural and forced convection coefficients are combined by cubing them and taking the cube root.

Equipment Capacity
The equipment capacity is added to the heat balance for the supply duct air (Node 15). The capacity includes the waste heat from the air handler. Currently this waste heat is a required input as there is no air handler performance model in REGCAP.

The capacity, the energy efficiency ratio (EER) and the power consumption (ratio of the capacity and EER) vary with the refrigerant charge, the coil temperature and the air handler flow. This model combines (Proctor.1999) with laboratory data from Texas A&M laboratory studies (Rodriguez et al. (1995)) to determine empirical correction factors that take into account the variation of incorrect charge of refrigerant as well as the temperature of the coil for three control types (capillary tube, orifice and thermostatic expansion valve (TXV)).

Refrigerant charge effects

In the following tables, CD is the charge deviation. So CD=-0.1 is a 10% undercharge.

Valve Type	Charge deviation capacity multiplier with a wet coil			
	CD<-0.316	**-0.316<=CD<=-0.15**	**-0.15<CD<=0**	**CD>0**
TXV	1+(1+CD-0.85)	1+(1+CD-0.85)	1	1
Cap Tube	0.4	1-6*CD^2	1-6*CD^2	1-CD*0.35
Orifice	0.4	1-6*CD^2	1-6*CD^2	1-CD*0.35

Valve Type	Charge deviation EER Multiplier with a wet coil			
	CD<=-0.15	**-0.15<CD<=-0.1**	**-0.1<CD<=0**	**CD>0**
TXV	1+(1+CD-0.85)*0.9	1	1	1-CD*0.35
Cap Tube	1+(1+CD-0.9)*1.35	1+(1+CD-0.9)*1.35	1	1-CD*09
Orifice	1	1	1	1-CD*0.25

Figures 9 and 10 represent the measured data from Rodriguez and the model in the above tables. The “old” model was based on a previous analysis of laboratory data and is not currently used in REGCAP.

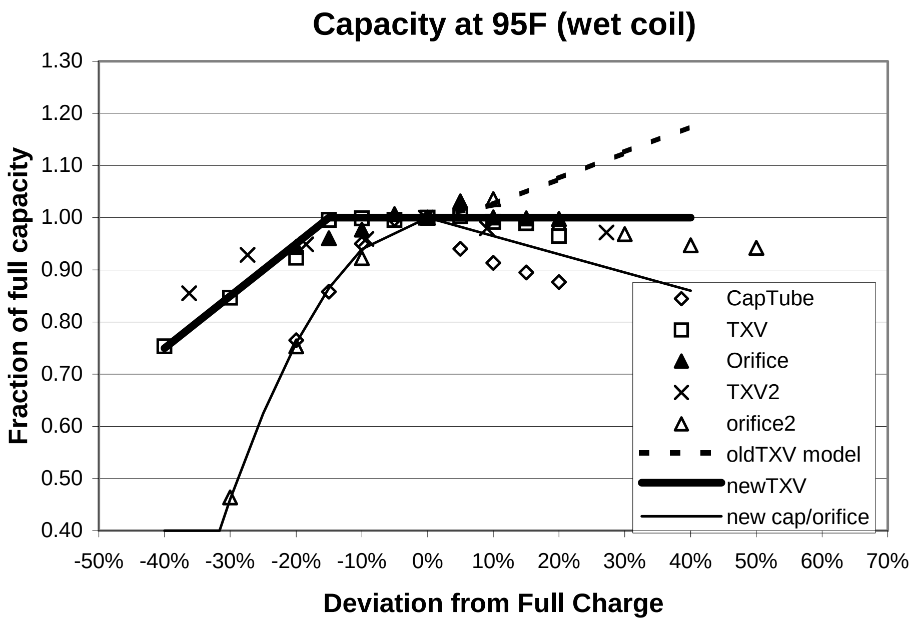

Figure 9. Wet Coil capacity variation with charge

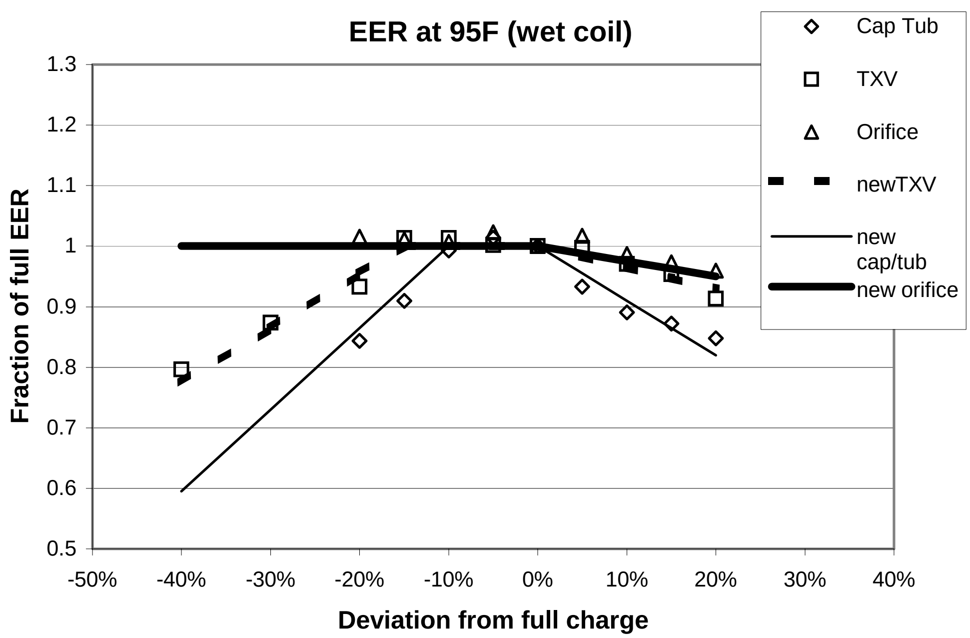

Figure 10. Wet Coil EER variation with charge

Valve Type	Charge deviation capacity multiplier with a dry coil		
	CD<-0.2	**-0.2<=CD<0**	**CD>=0**
TXV	1.2+CD	0.925	0.925
Orifice/cap tube	0.94+CD*0.85	0.94+CD*0.85	0.94-CD*0.15

Valve Type	Charge deviation EER multiplier with a dry coil	
	CD<0	**CD>=0**
TXV	1.04+CD*0.15	1.04-CD*0.35
Orifice/cap tube	1.05+CD*0.5	1.05-CD*0.35

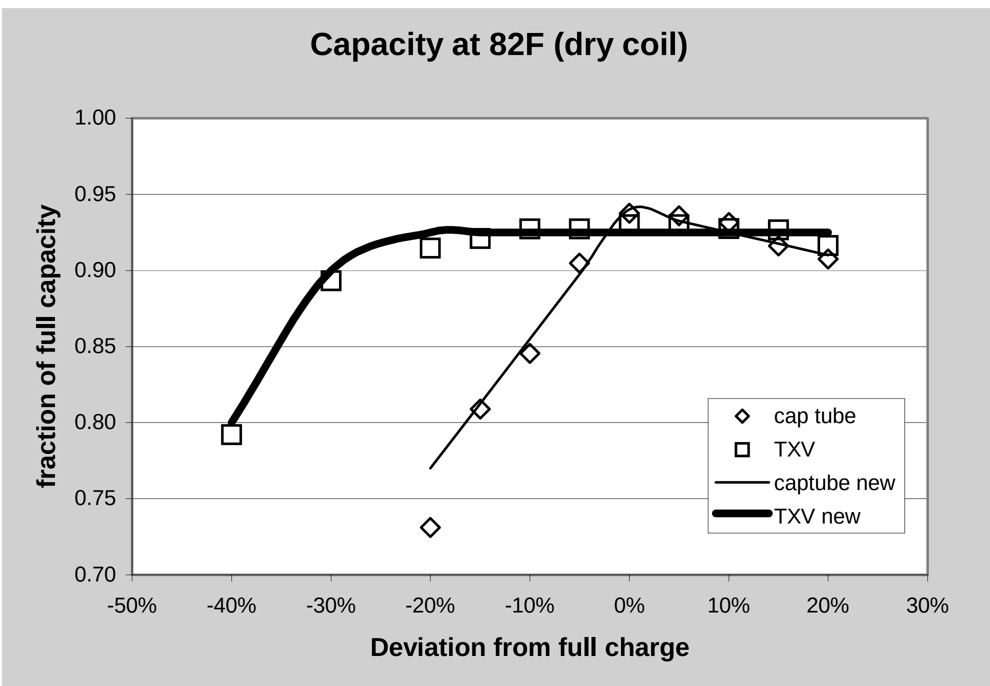

Figure 11. Dry Coil capacity variation with charge

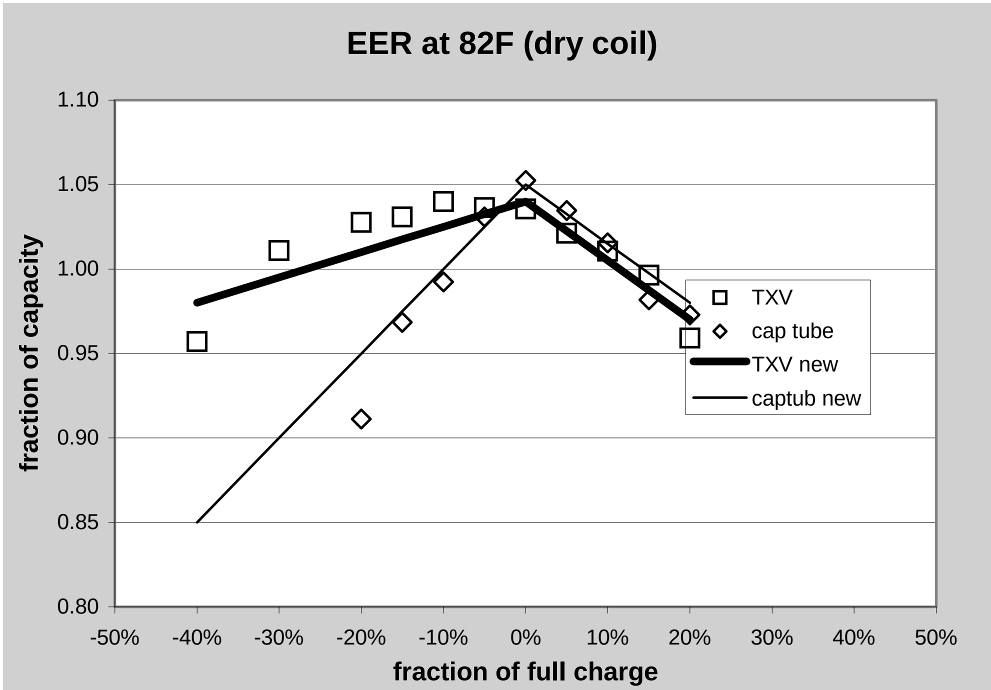

Figure 12. Dry Coil EER variation with charge

Outdoor air temperature

The outdoor air corrections are relative to the reference temperatures used for rating:

- Capacity:
  Correction = (-0.00007)*(T-Tref)^2-0.0067*(T-Tref)+1
- EER:
  Correction = (-0.00007)*(T-Tref)^2-0.0085*(T-Tref)+1

Tref is 95F (35ºC) for a wet coil and 82F (28ºC) for a dry coil.

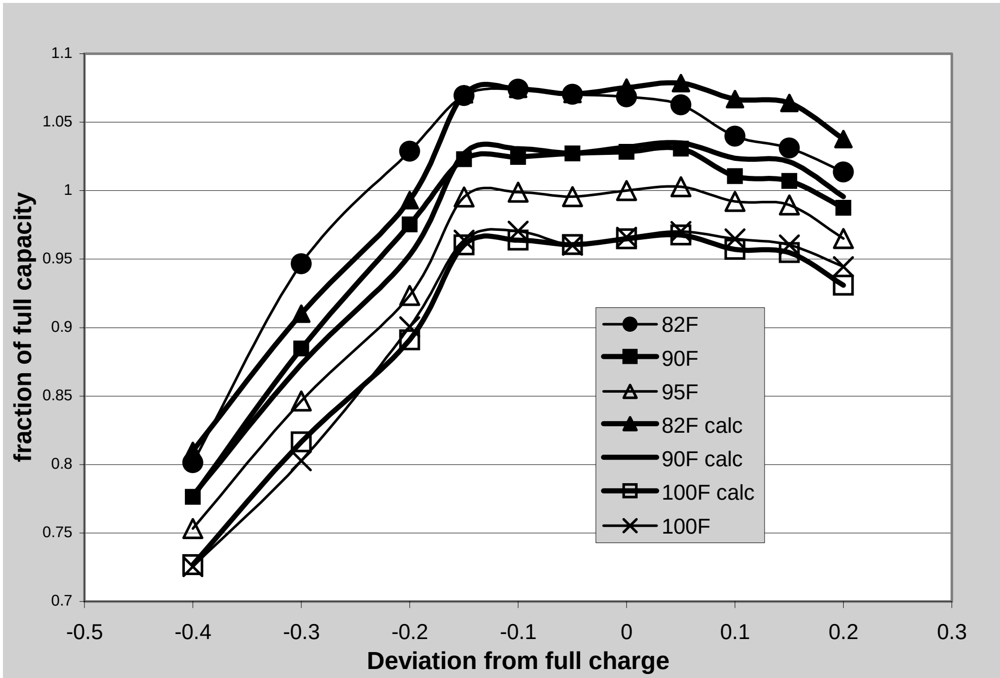

Figure 13. Dry Coil capacity variation with outdoor temperature

Air handler flow

The same multiplier is used for capacity and EER. These are taken from ASHRAE standard 152 and were developed for the standard by John Proctor from correlations to Texas A&M laboratory data.

- for TXV:

$$correction = 1.62 - 0.62\left(\frac{Qactual}{Qrecommended}\right) + 0.647\ln\left(\frac{Qactual}{Qrecommended}\right)$$

- for capillary tube and orifice:
- $correction = 0.65 + 0.35\left(\frac{Qactual}{Qrecommended}\right)$

Qrecommended is the airflow recommended by the manufacturer – typically 350 to 400 cfm/ton.

Node Heat Transfer Equations

In each of the following equations the subscript on temperature, T, refers to the node location and the superscript to the timestep.

Node 1. Attic Air

The attic air has convective (the h_U terms) heat transfer from all the interior attic surfaces - nodes 2, 4, 8, 6 and 9 as shown Figure 7. Although each convection term uses the same velocity, U_U, the different temperatures will change the film temperature, T_ε, and thus the heat transfer coefficient. In addition the convective flows in and out of the attic, M_a, and the flow through the ceiling, M_c, duct leakage and duct leak air handler off flows transport heat in and out of the attic air.

Nodes 2,3,4,5,9 and 10

These nodes all experience internal conduction with surface convection and radiation. The differences are that the exterior sheathing surfaces have daytime solar gains and nightime radiation cooling.

The areas of nodes 3 and 5 (exterior surfaces) are increased by 50% for tile roofs.

Node 6. Attic Joists and Trusses

The joists and trusses only exchange heat with the attic air by convection.

Nodes 7 and 8. House Ceiling/Attic Floor

The underside of the ceiling has radiant exchange with the inside surfaces of the house that are assumed to be at T_{in}, i.e. the same temperature as the air in the house. The house is assumed to have internal free convection and so the ceiling exchanges heat with the house air. There is also conduction through the ceiling to the floor of the attic.

The attic floor exchanges heat by radiation to the pitched roof surfaces, forced convection with the attic air and by conduction through the ceiling form the house below. The radiation terms are important because during high daytime solar gains the warm sheathing can raise the attic floor temperature above the attic air and reduce heat loss through the ceiling. Conversely cooler attic sheathing on clear nights will make the attic floor colder.

Node 11. Return duct external duct surface

Exchanges heat by convection plus the thermal resistance of the duct walls with the return duct air, by convection with the attic air and radiation with attic surfaces.

Node 12. Return Duct Air

The return duct with air handler on has air entering at indoor temperature plus leakage at attic temperature and air leaving at the air handler flow rate. There is also forced convection plus the thermal resistance of the duct walls between the return duct air and the return duct surface. With the air handler off the processes are the same but the air flow rate is determined by the leakage area of the duct leaks.

Node 13. House mass

The thermal mass of the house is an empirical approximation based on assuming that the first 5 cm of the concrete slab and 1 cm of the drywall all interact with the attic air. The surface area for heat transfer for the house thermal mass has been empirically adjusted to be 2.5 times the wall and floor surface area. 95% of the solar gain to the house (calculated from the direct and diffuse solar radiation, solar geometry and window area) goes to the thermal mass. The other 5% goes to the house air.

Node 14. Supply duct external duct surface

Exchanges heat by convection plus the thermal resistance of the duct walls with the supply duct air, by convection with the attic air and radiation with attic surfaces.

Node 15. Supply Duct Air

The supply duct with air handler on has air entering at the return temperature (at the air handler flow rate) and air leaving through leaks to the attic and also to the house. There is also forced convection plus the thermal resistance of the duct walls with the duct surface. With the air handler off the processes are the same but the air flow rate is determined by the leakage area of the duct leaks. The equipment capacity is added to the supply duct air (noting that cooling capacity is negative).

Node 16. House Air

House air exchanges energy by convection with the ceiling and the house internal mass. Air flows due to inflows and outflows through the envelope and register grilles are included. Care must be taken to ensure that the appropriate mass fluxes are used when the air handler is on or off and that the flow directions are tracked (particularly for the ceiling and duct air handler off flows) so that the correct air temperature is used for each air flow. The solair temperature is used together with the envelope UA to calculate the heat transfer through the house envelope. Solar loads are dealt with by having 5% of the solar gain go to the air in the house and the other 95% to the house mass. The solar gain is through windows only and includes a shading coefficient and the solar gain through the windows in each of the four cardinal directions. Any internal loads go directly to the house air.

Envelope load

$$Load = UA\left(t_{solair} - t_{in}\right) + 0.05 q_{solgain} \tag{55}$$

where $q_{solgain}$ is the average solar radiation on the walls over four cardinal directions and includes any shading, and

$$t_{solair} = t_{out} + 0.03 q_{incidentsolar} \tag{56}$$

$q_{incidentsolar}$ is the average incident solar radiation on each vertical surface for the four cardinal directions.

The factor 0.03 is from ASHRAE Fundamentals SI p. 26.5 (1993).

Solution of the Attic Heat Transfer Equations

At each node the rate of change of thermal energy is equated to the sum of the heat fluxes due to radiation, convection and conduction. This results in the above set of equations that are linear in temperature and must be solved simultaneously. This simultaneous solution is found using Gaussian elimination. When the temperatures have been calculated the attic air temperature (Node 1) is returned to the attic ventilation model so that a new attic ventilation rate can be calculated. This new ventilation rate is then used in the thermal model at the attic air node to calculate temperatures. This iterative process is continued until the attic air temperature changes by less than 0.1^{o}C. Because the attic ventilation rates are relatively insensitive to the attic air temperature usually fewer than five iterations between thermal and ventilation models are required.

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Appendix B: Latitude and altitude taken from ACM joint Appendix

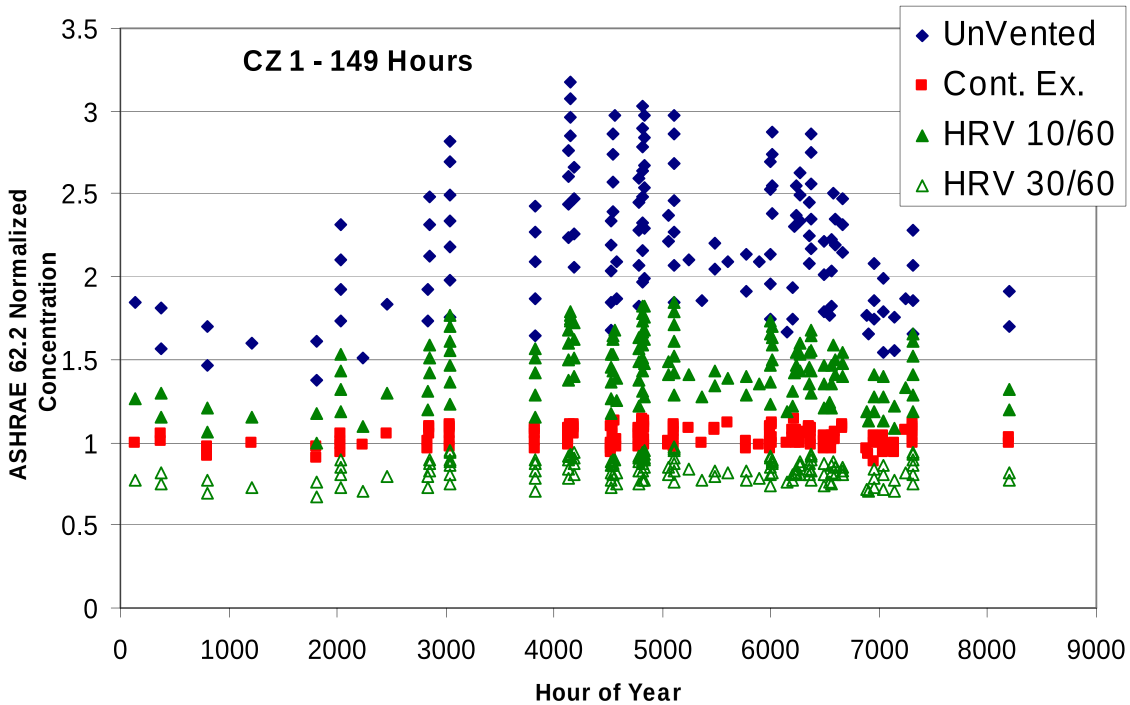
Table II-1 –California Climate Zone Summary

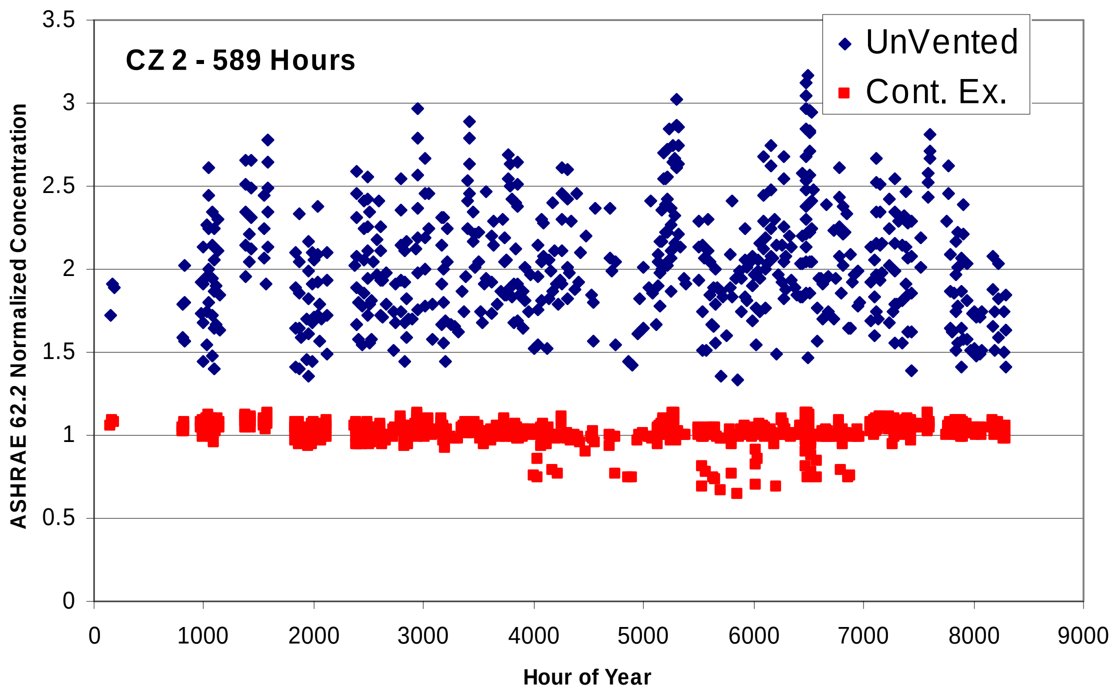
Climate Zone	City	Latitude	Longitude	Elevation
1	Arcata	40.8	124.2	43
2	Santa Rosa	38.4	122.7	164
3	Oakland	37.7	122.2	6
4	Sunnyvale	37.4	122.4	97
5	Santa Maria	34.9	120.4	236
6	Los Angeles AP	33.9	118.5	97
7	San Diego	32.7	117.2	13
8	El Toro	33.6	117.7	383
9	Burbank	34.2	118.4	655
10	Riverside	33.9	117.2	1543
11	Red Bluff	40.2	122.2	342
12	Sacramento	38.5	121.5	17
13	Fresno	36.8	119.7	328
14	China Lake	35.7	117.7	2293
15	El Centro	32.8	115.6	-30
16	Mt. Shasta	41.3	122.3	3544

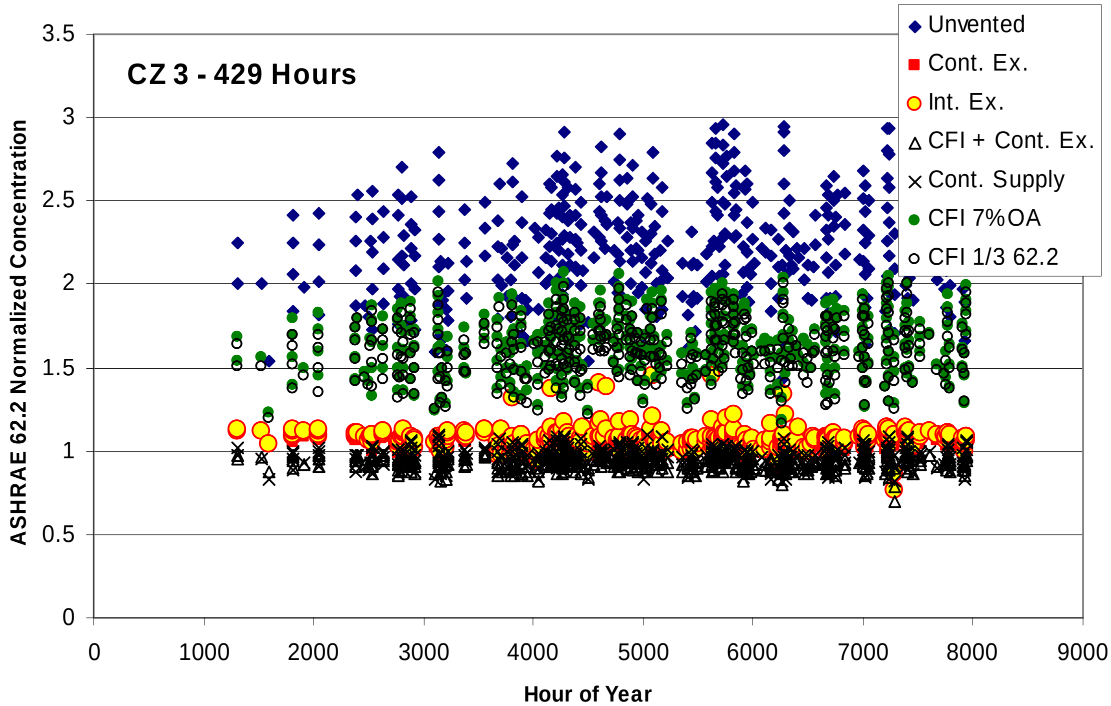
Appendix C: Summary of Sizing based on Chitwood Field data

Location		Cooling Sizing	Heating Sizing	Cooling/Heating Ratio
		tons/1000ft^2	kBtu/1000ft^2	tons/100kBtu
ALL CZ	average	2.2	45.8	4.9
	max	5.0	113.2	7.5
	min	0.8	20.1	2.8
	sdev	0.7	16.9	1.0
	sdev%	31.9	37.0	19.8
CZ11	average	1.8	39.3	4.6
	max	2.6	56.3	6.0
	min	0.8	20.1	3.2
	Sdev%	33.6	31.1	13.7
CZ12	average	1.6	40.5	3.9
	Max	2.0	55.5	5.0
	Min	1.1	30.9	2.8
	Sdev%	22.0	21.2	20.9
CZ8	average	2.0	32.6	6.5
	Max	2.1	45.7	7.5
	Min	1.8	24.2	4.4
	Sdev%	7.2	35.1	27.9
CZ15	average	2.9	61.7	4.9
	Max	5.0	113.2	5.8
	Min	2.2	38.0	3.0
	Sdev%	25.7	37.4	15.9
CZ10	average	2.1	33.0	6.3
	Max	2.1	35.1	6.7
	Min	1.9	31.9	5.6
	Sdev%	5.2	5.4	10.2
CZ13	average	2.3	46.6	5.0
	Max	2.6	58.7	6.0
	Min	1.5	26.2	4.0
	Sdev%	13.2	19.2	15.4

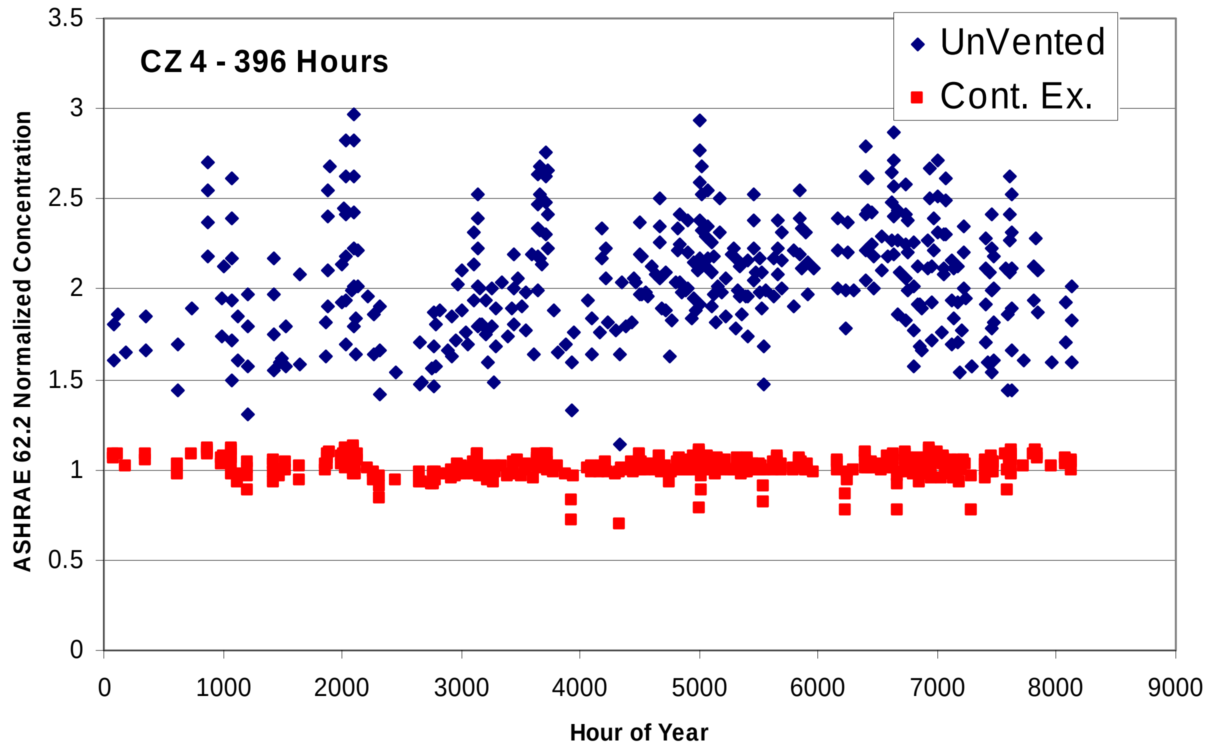
Appendix D. Results of low ventilation rate indoor concentration calculations



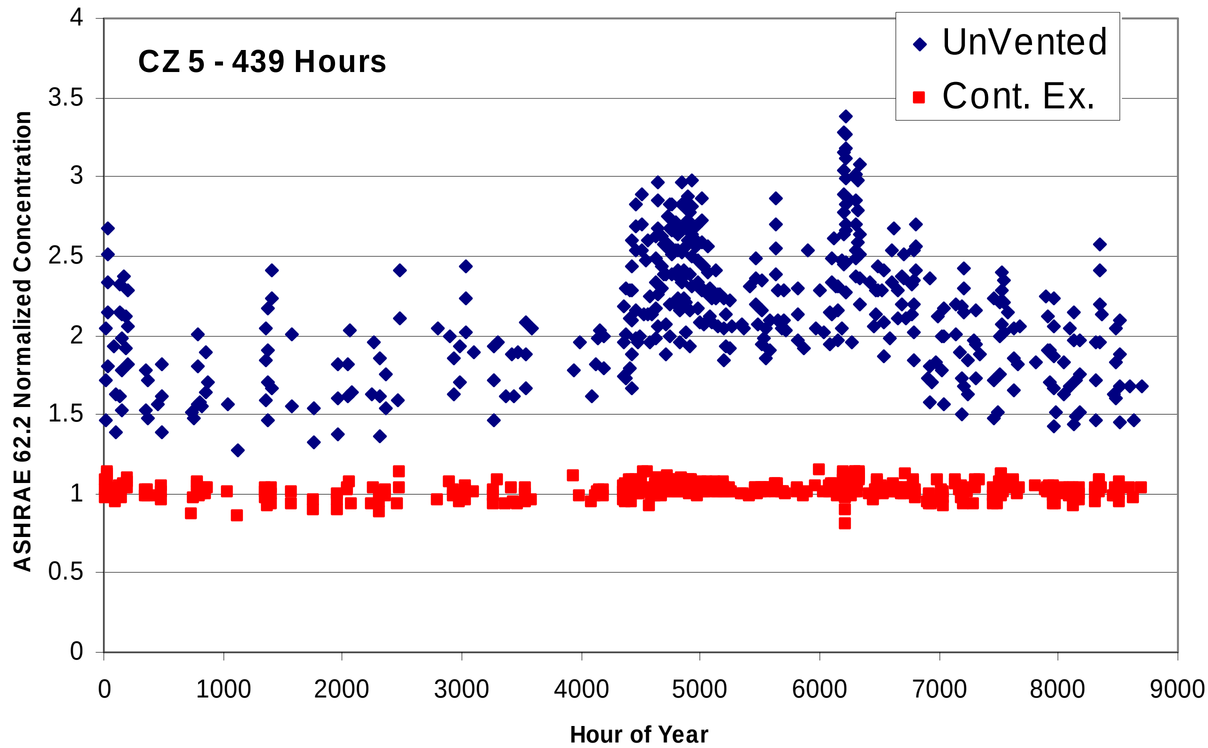




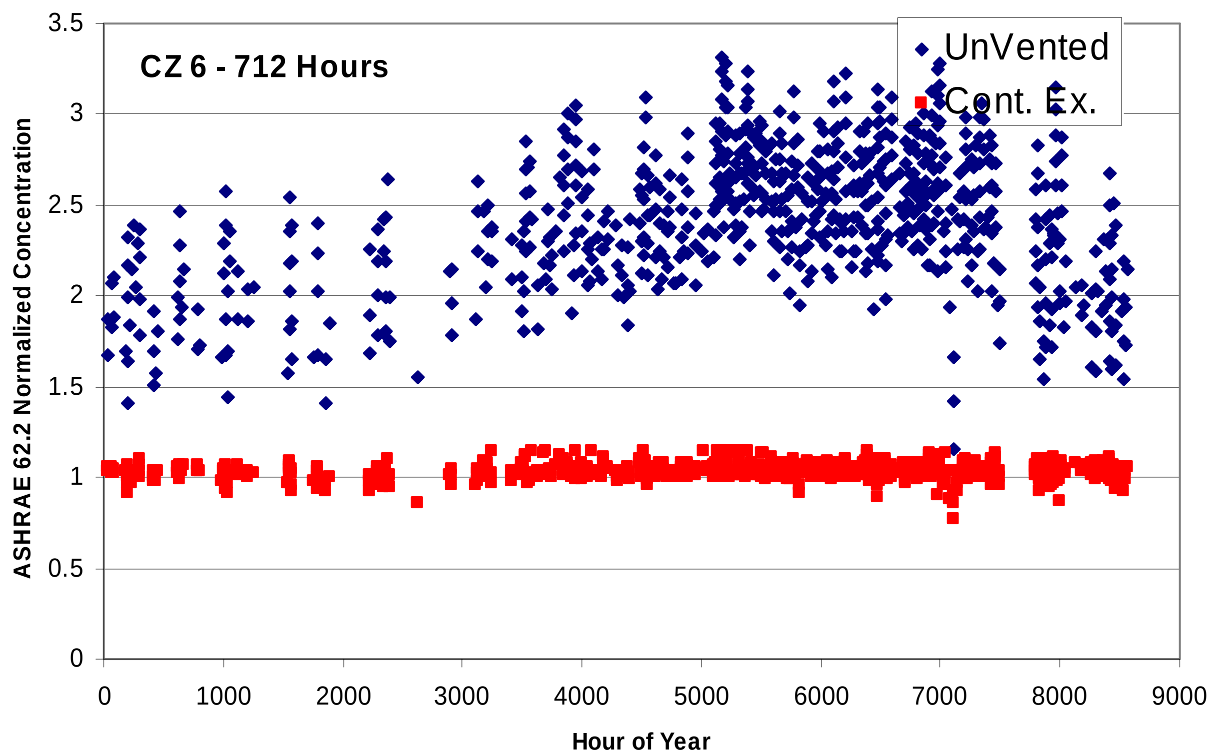
CZ 3 - 429 Hours
Unvented
Cont. Ex.
Int. Ex.
CFI + Cont. Ex.
Cont. Supply
CFI 7%OA
CFI 1/3 62.2
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000



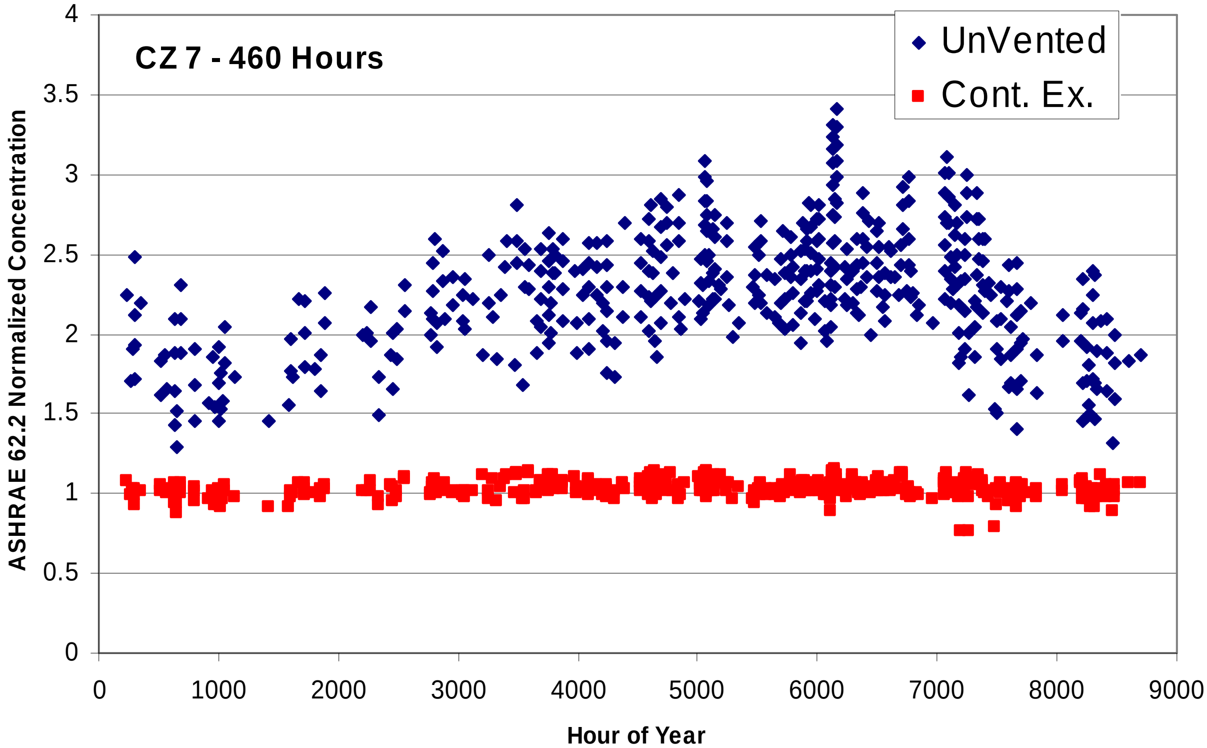
CZ 4 - 396 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

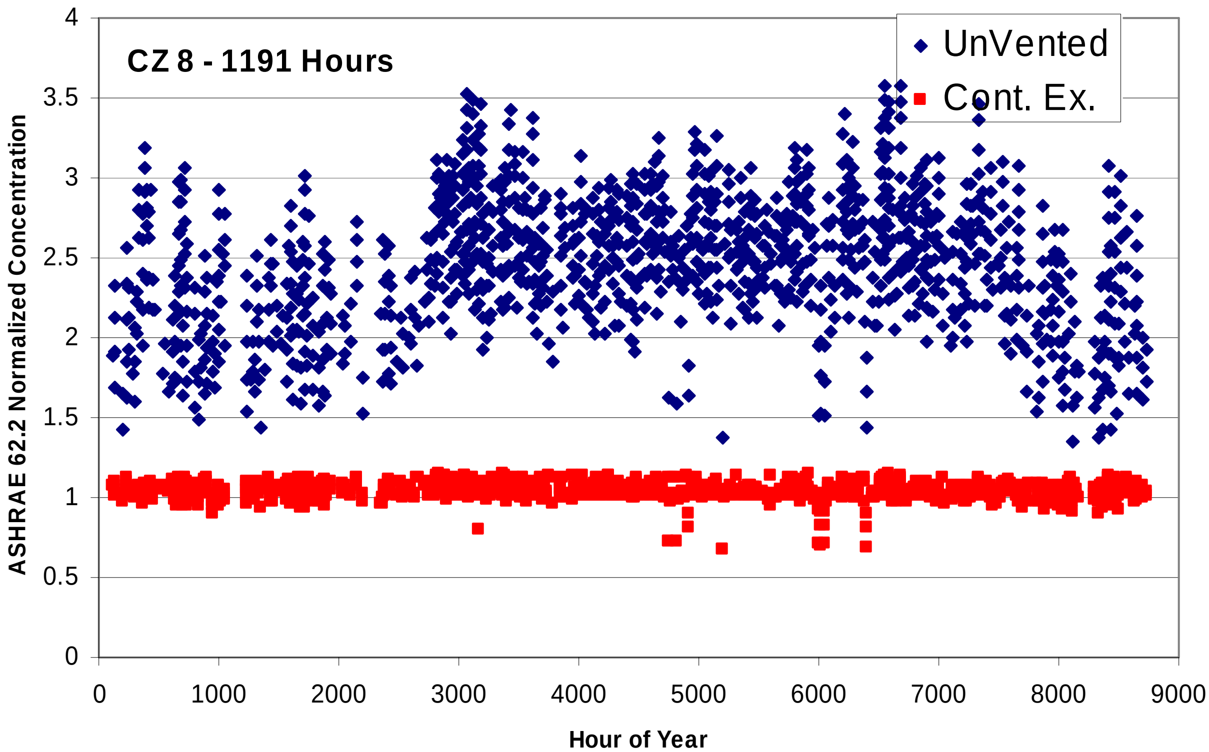


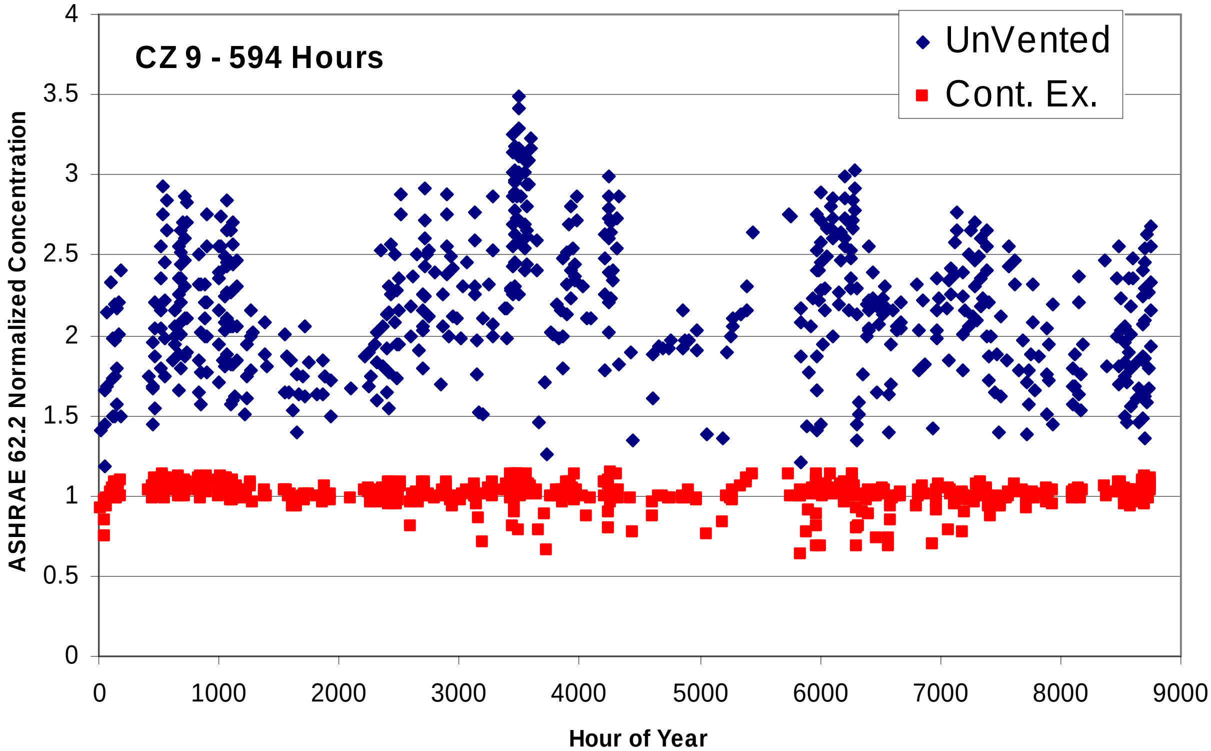
CZ 5 - 439 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

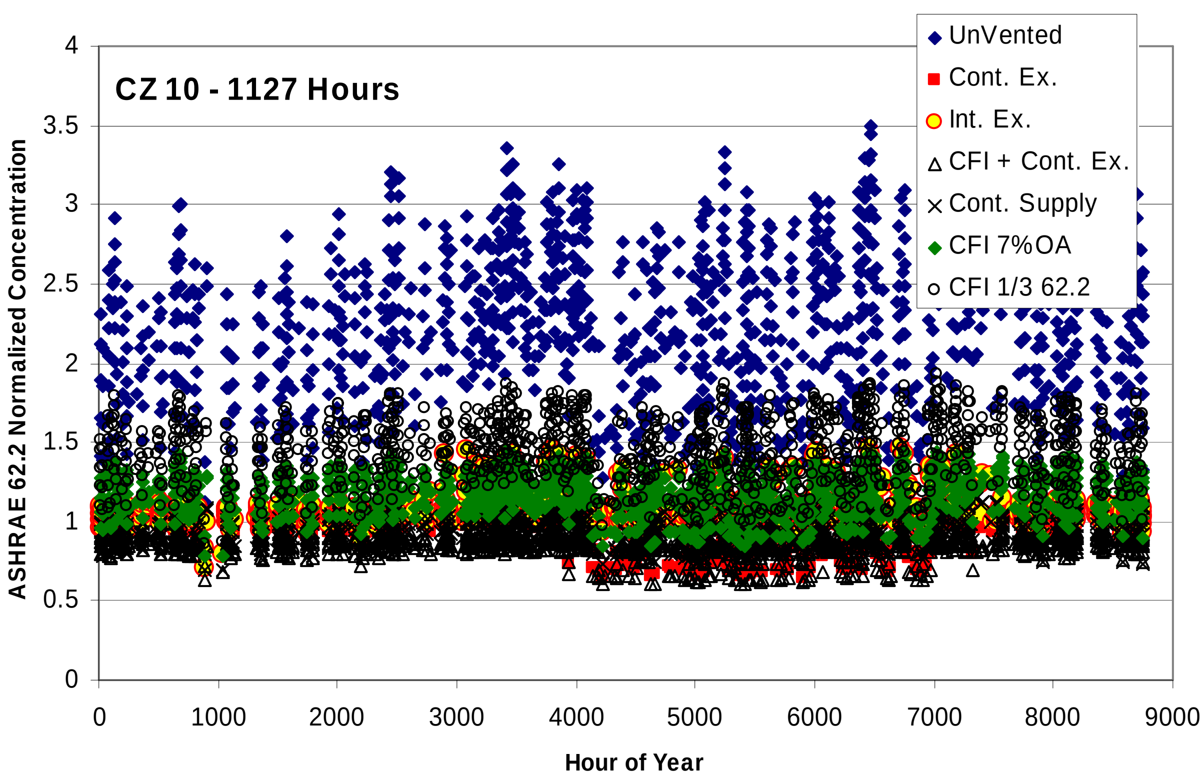


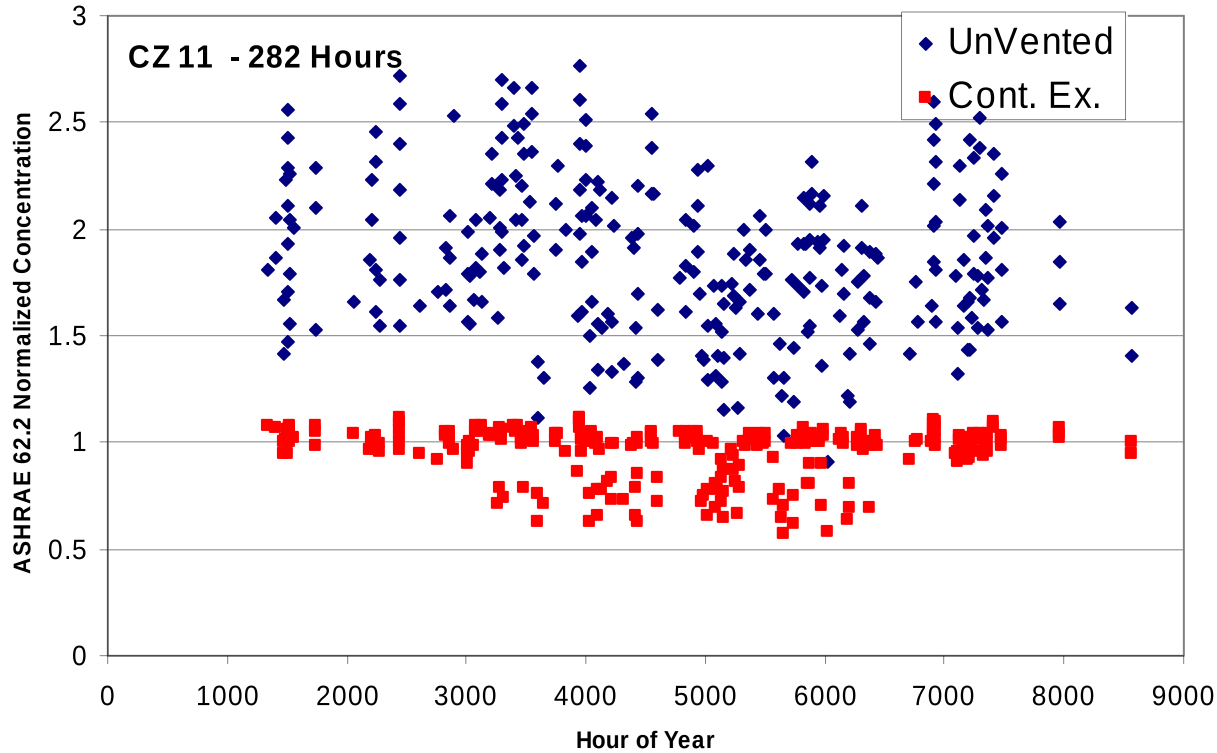
CZ 6 - 712 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 7 - 460 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

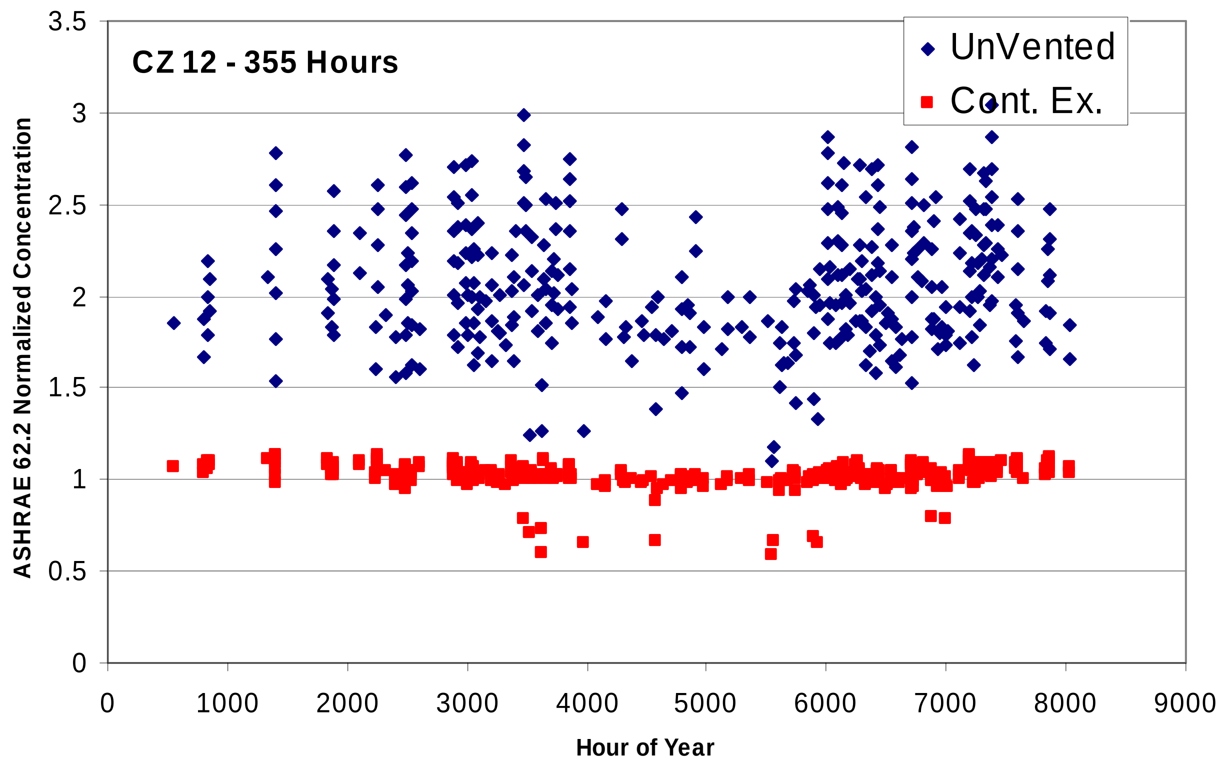

CZ 8 - 1191 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 9 - 594 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

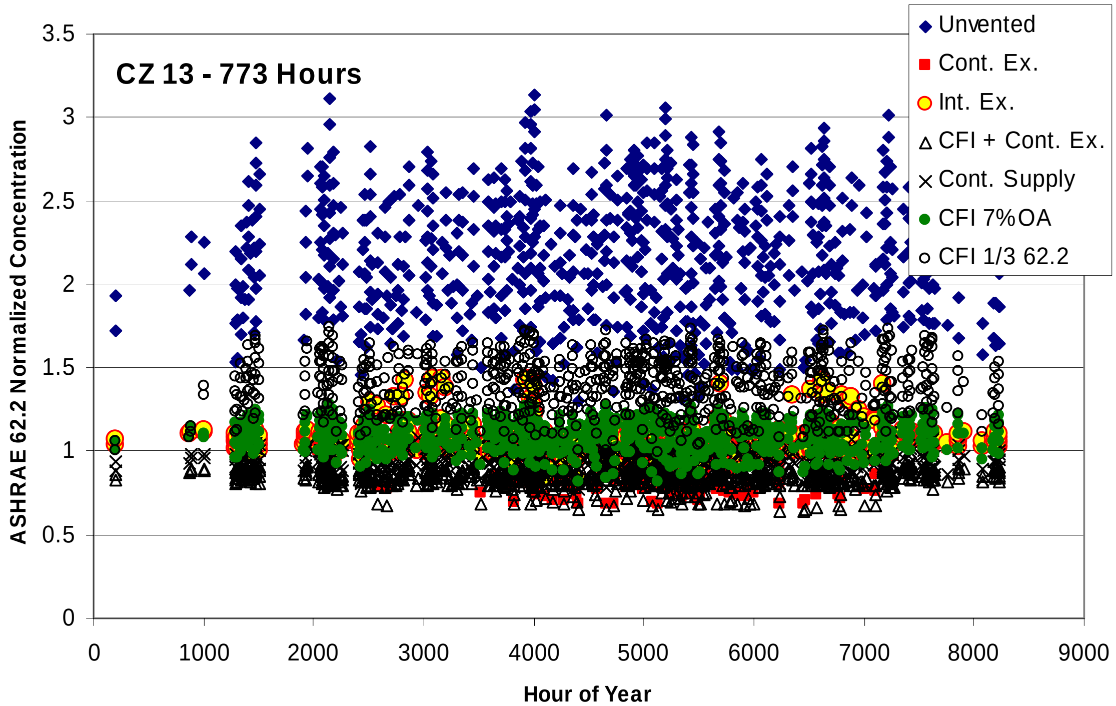

CZ 10 - 1127 Hours
UnVented
Cont. Ex.
Int. Ex.
CFI + Cont. Ex.
Cont. Supply
CFI 7%OA
CFI 1/3 62.2
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

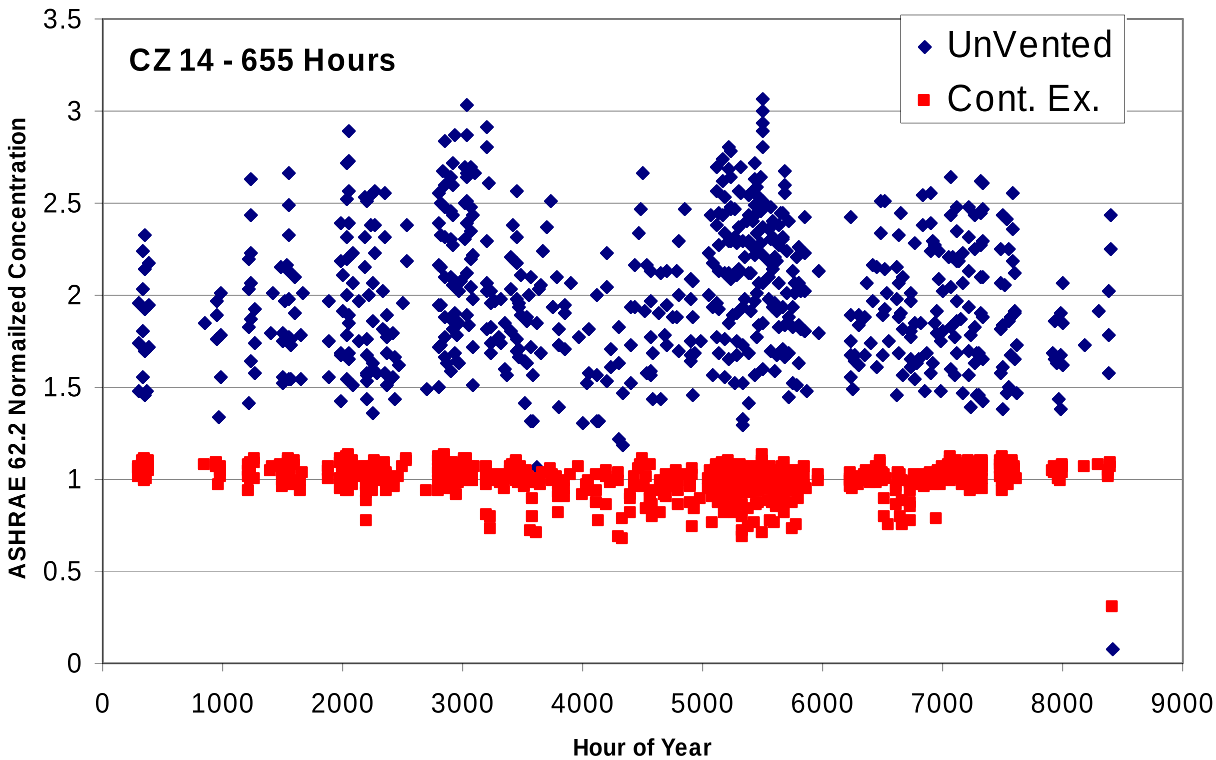


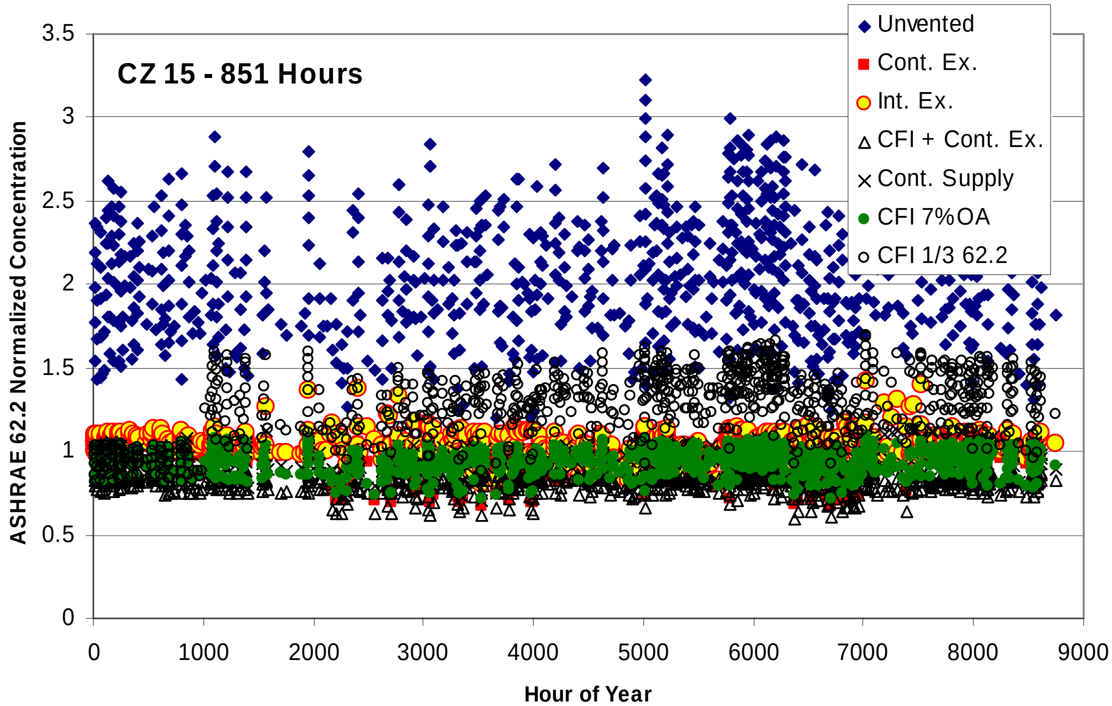
CZ 11 - 282 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

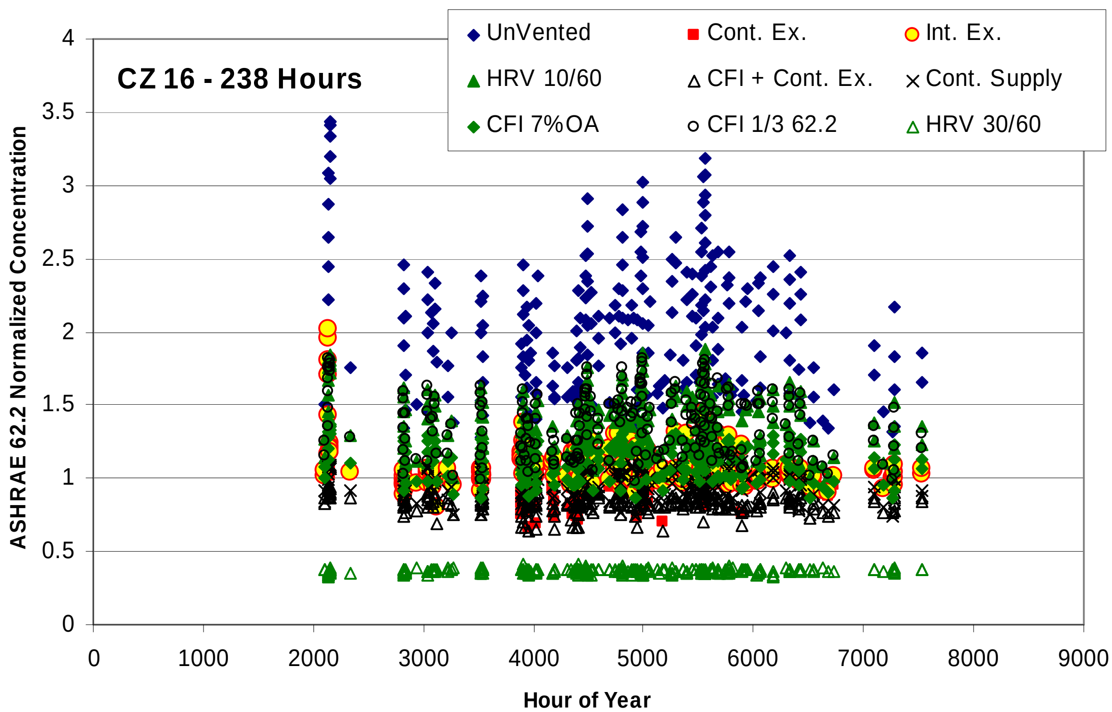


CZ 12 - 355 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 13 - 773 Hours
ASHRAE 62.2 Normalized Concentration
Hour of Year
Unvented
Cont. Ex.
Int. Ex.
CFI + Cont. Ex.
Cont. Supply
CFI 7%OA
CFI 1/3 62.2
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 14 - 655 Hours
ASHRAE 62.2 Normailzed Concentration
Hour of Year
UnVented
Cont. Ex.
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 15 - 851 Hours
Unvented
Cont. Ex.
Int. Ex.
CFI + Cont. Ex.
Cont. Supply
CFI 7%OA
CFI 1/3 62.2
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 16 - 238 Hours
UnVented
Cont. Ex.
Int. Ex.
HRV 10/60
CFI + Cont. Ex.
Cont. Supply
CFI 7%OA
CFI 1/3 62.2
HRV 30/60
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

Appendix E. Equipment Capacity and Blower Power Consumption

The equipment capacity was based on the results of a field survey of 60 new California houses performed as part of another PIER study (Rick Chitwood). The resulting heating and cooling capacities are generally greater than those estimated using sizing calculations such as ACCA Manual J/S procedures. In some cases the cooling capacity deterimes the heating capcity due to the limited packaging alternatives that are commercially available. Primarily this is an issue of furnace blower motor operating ranges that restrict the differences in heating and cooling capacities that can be serviced by an individual blower.

Climate Zone	Heating Capacity (KBtu/h)	Cooling Capacity (Tons)	Heating Blower Power (W)	Cooling (and Ventlating) Blower Power (W)
1	94	1.5	630	300
2	97	4	655	800
3	84	1.5	563	300
4	84	2	563	400
5	61	3.5	412	700
6	61	3.5	412	700
7	57	4	386	800
8	72	3.5	487	700
9	87	4	588	800
10	73	3.5	487	700
11	87	3	580	600
12	73	3	596	600
13	103	4	689	800
14	107	5	722	1000
15	136	5	916	1000
16	147	3.5	983	700